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Torque loss in helical gears: Influence of "low loss" gear design

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Abstract

Over the years and, with the scarcity of some energy sources, power losses generated in mechanical transmissions have become an important focus of study. This study was focus on the decrease of energy consumption using a new gear geometry designed with the purpose to promote a considerable reduction in friction between contacting teeth.

The main objective of this work was the study of the influence of different oil formulations and of different tooth gear geometries in the torque loss.

The study involved testing of eight different oil formulations, being four based on Polialphaolefins (PAOR, PAOC, PAOX, PAOM), two based on Mineral oil (MINR, MINE), one Ester (ESTR) and a Polyallkylene Glycol (PAGD). All the selected oils were tested on three different gear geometries, two of them using the "low loss" concept (501, 951) and a standard spur type-C geometry (C40). This test matrix will allow to studying the influence of the combination gear design/ oil formulations on the torque loss.

The tests were performed in the FZG test rig, at low and high speeds (200 rpm, 400 rpm and 1200 rpm) and different torques (3.3 N.m, 69.99 N.m, 132.48 N.m and 215.57 N. m), in order to simulate realistic operating conditions. During all the tests performed, the operating temperatures and the torque loss were continuously measured. The oil injection temperature was controlled and maintained at 80°C (±1°C).

A numeric model was used in order to predict and quantify the different sources of power losses inside the gearbox. The calibration of the numerical model was made using the experimental data obtained on the experimental tests.

Resumo

Com o passar dos anos e, com a escassez de algumas fontes de energia, as perdas de potência geradas nas transmissões mecânicas tornaram-se um importante foco de estudo. Para tal, o estudo foca-se na diminuição do consumo de energia com o aumento da eficiência. Recentemente, uma nova geometria de engrenagens foi projetada com o intuito de permitir uma diminuição considerável no atrito entre os dentes engrenados.

O principal objetivo deste trabalho é o estudo da influência dos óleos e das diferentes geometrias de engrenagens na perda de binário.

Com o intuito de estudar estas necessidades, por intermedio de ensaios e análises de resultados, foi realizada a comparação de oito óleos, sendo quatro Polialfaolefinas (PAOR,PAOC,PAOX,PAOM), dois Minerais (MINR,MINE), um Ester (ESTR) e um Polialquilenoglicol (PAGD). Além da comparação entre óleos também foram comparadas três diferentes geometrias de engrenagens sendo duas “low loss” (501, 951) e uma do tipo-C (C40). Os testes foram efetuados numa FZG, a baixas e altas velocidades (200 rpm, 400 rpm e 1200 rpm) e a diferentes binários (3.3 N.m, 69.99 N.m, 132.48 N.m e 215.57 N.m) de maneira a garantir as condições reais. As temperaturas e as perdas de binário foram medidas durante o decorrer de todos os testes. A temperatura de injeção foi controlada ao longo de todos os ensaios e mantida a 80°C.

Foi usado um modelo numérico com o intuito de prever e quantificar as diferentes perdas de potência no interior da caixa de testes. A calibração do modelo numérico foi feita com o auxílio aos dados experimentais obtidos.

Keywords

Wind turbine gear oils
FZG test rig
Power loss
Efficiency
Coefficient of friction
Polialphaolefin oil
Mineral oil
Ester oil
Polyallkylene Glycol oil
Meshing torque loss
Churning loss
Rolling Bearings Power loss

Palavras chave

Lubrificantes para engrenagens multiplicadoras de aerogeradores
Banco de ensaios FZG
Perda de Potencia
Eficiência
Coeficiente de atrito
Óleo Polialfaolefinas
Óleo Mineral
Óleo Ester
Óleo Polialquilenglicol
Perdas de potência por atrito nos engrenamentos
Perdas de potência por chapinagem
Perda de potência nos rolamentos

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1 Introduction

Gearboxes efficiency is directly linked to a higher energy consumption. In order to reduce the energy consumption, as the energy resources are increasingly smaller, the necessity to evaluate and improve the efficiency is imposed.

Efficiency increase is achieved by decreasing the power loss. The “low loss” gear design was developed in order to reduce the meshing power losses.

The study of the lubricant properties and the performance of each oil formulation with different gear geometries lead to the best combination for the pretending conditions.

1.1 Aim and thesis outline

This project is the result of the experimental and numerical work accomplished for the course of Master's Degree on Mechanical Engineering under the branch of Project and Mechanical Construction and was realized at CETRIB (Tribology, Industrial Maintenance and Vibrations Unit of INEGI).

This thesis is composed by 6 chapters, including this introductory chapter.

Chapter two is dedicated to present the different lubricant formulations, their tribological properties and the additive packages used to improve the performance of each base oil type.

In Chapter three, the test rig used on this work is described. All the materials and the test procedures are also included on this chapter.

The Fourth Chapter is dedicated to the experimental results. In this chapter the experimental results of torque loss and temperatures, as well as the efficiency of the test gearbox are presented and discussed.

Chapter Five analysis the different power loss sources occurring within the gearbox. A power loss model was developed in order to study and validate the experimental results.

The finally chapter is dedicated to the conclusions of the realized work and to the future work.

2 Lubricants and their properties

2.1 Introduction

The lubricants perform different functions but the most important one is to minimize the coefficient of friction between contacting surfaces with relative motion.

Over the years, the demand made on lubricants has increased, both technologically and environmentally requiring an increased capacity to minimize wear, evacuate the heat from contact zone, support higher operating temperature, be miscible with additive package and others chemicals and minimize the environmental impact. Lubricating oils are a mixing of an base oil and an additive package in order to meet these requirements.

All subjects treated in this paragraph are based in [1].

2.2 Lubricants types

The world of lubricants presents a wide range of choice in function of the user requirements. Each lubricant presents different categories of base oils and additive packages. Several lubricant types are available: liquid lubricants, greases, solid lubricants and gaseous lubricants.

2.2.1 Lubricant oils

The lubricating oils are the most common and have different origins such as: vegetable, animal, petroleum or synthetic.

2.2.2 Vegetable and animal based oils

The vegetable and animal oils were the first lubricants used, but due to their chemical inertness and with the increasing of requirements are replaced by mineral and synthetic oils.

The major advantages of vegetable oils when compared with the minerals oils are the higher VI, low rate of evaporation and, the more important point, the fast biodegradability. On the other hand, they oxidize quickly, showing low resistance to high temperatures.

Nowadays more and more vegetable base oils are available because the environmental requirements.

2.2.2.1 *Mineral based oils*

The petroleum base oils or mineral based oils are composed by natural hydrocarbons resulting by decomposition of organic wastes. These are classified by three broad categories: Paraffinic, Naphethenic and Aromatic base oils.

Paraffinic bases are the most used and exhibit a low specific weight, freeze point, great oxidation resistance, low viscosity variation with temperature and therefore a high viscosity index. Furthermore, paraffinic oils are also elastomer friendly (elastomer are used to build seals). Nevertheless, the high molecular weight can lead to crystallization of the oil at ambient temperature.

The use of naphethenic bases is limited to applications with small thermal variation. They are characterized by their high specific weight, high freeze point and small resistance to oxidation. Unlike the paraffin basis, these present a high viscosity variation with temperature, therefore a low viscosity index, and are relatively aggressive with elastomers.

Aromatic bases are normally unwanted because they represent a small proportion of mineral oils. Real base oils are, in general, a combination of naphthenic basis and paraffin bases. This mixture is used in order to combine different properties.

2.2.2.2 *Synthetic based oils*

The synthetic based oils are created by synthetized hydrocarbons, exhibit different bases such as base of petroleum products, vegetable products or others. Synthetic bases present favorable properties compared to other bases, but they have a higher cost. Compared with mineral bases, synthetic bases exhibit better performance especially in oxidation resistance, in Viscosity Index and in friction. Synthetic hydrocarbons, polyglycols, esters and silicones are the most distinctive.

2.3 Additive package

A lubricant is composing by base oil and an additive package, in order to obtain the desired properties. Nowadays, almost all lubricants types have aggregated with then an additive package (at least one). The volume of additives in the final product varies between few hundredths up to 30%. With the help of an additive package, significant progress has been obtained in developing in internal combustion regimes and industrial machinery.

Viscosity Index additives is used to increase the Viscosity Index, in the other words, decrease viscosity at lower temperatures and/or increase at high temperatures.

The Viscosity Index is improved with addiction of high molecular weight polymers. With the addiction of polymers oil viscosity becomes higher at high temperatures.

This additive is used in engine oils, automatic gearbox oils and hydraulic systems.

Anti-Wear (AW) and Extreme Pressure (EP) additives are used to reduce friction and wear in extreme conditions of lubrication. These can be lubricity agents, anti-wear additives and extreme pressure additives.

Lubricity agents are used to decrease fiction on limit conditions of lubrication. It is noteworthy that the amount and the type depend on the application. This additive is used to oppose, for example, stick-slip phenomena.

Anti-wear additives create a protection film, between the contacting surfaces,through a chemical reaction with metallic surfaces thereby enhancing lubricant anti-wear properties. Some examples: fatty oils, organic compounds and phosphoric esters.

Extreme Pressure additives are used in order to avoid adhesion between two metallic surfaces in conditions of extreme pressure and relative sliding. These additives create a protective film by chemical combination with metallic surfaces. This film reduces the possibility of occurring failure phenomena due the rupture of lubricant film caused by high contact pressure or high sliding speed. Nowadays, exist a high variety of extreme pressure additives are available. Adding them to the oil for gears meets all operational requirements. The extreme pressure additives are chemically very active therefore it is necessary to obtain a higher thermal stability and oxidation to avoid instability of the lubricant.

Antioxidants avoid, modify or retard the reaction of hydrocarbons (oil belonging to) with oxygen. Oxidation generates soluble acid compounds. These acids increase the viscosity and can render the oils corrosive leading to the formation of sludge and varnishes that can adhere to mechanical organs. The antioxidant can act on the oil or on the metallic surface in contact. In order to obtain stable oil with good corrosion resistance it is necessary to do a careful refining and a good selection of the type and concentration of the antioxidant.

Detergents are used to avoid the formation of gummy deposits and lacquers in engines.

Corrosion inhibitors additives intended not only to avoid corrosion of metallic surfaces but also to act as antioxidants.

The rust inhibitors have a high polar attraction for metallic surfaces and, by physical or chemical action, formed a continuous film in metallic surfaces, protecting them against water.

Foam inhibitors additives avoid the foam formation in oil when it is subjected to great upheavals. Foam inhibitors are used in small proportion and therefore do not require special precaution.

2.4 Physical properties

To define the physical properties of lubricants it is necessary to know the structure of the base oil.

2.4.1 Viscosity

Oil viscosity is the resistance that the lubricant molecules offer to inner slide between them. This value can be calculated using *Newton formula* relative to the laminar flow between two surfaces, when one moves with speed V and the other is fixed.

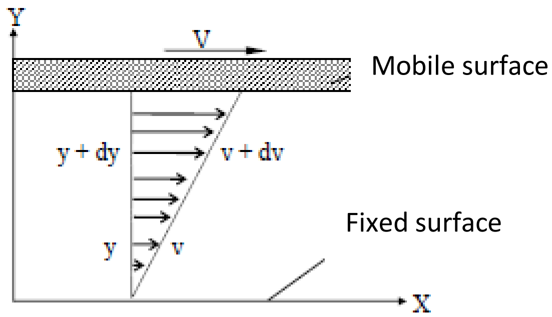

Figure 1: Laminar flow of a fluid

Looking at the relative movement between the two surfaces it is assumed that the fluid is divided in thin films. It is easy to understand that the different films move at different speed between 0 to V.

Establishing between the two surfaces a linear proportionality such that for a distance y of the fix surface the speed is v and for a distance (y + Δy) the speed is (v + Δv), then the sheer stress can be represented by equation (2.1).

$$\tau = \sigma_{xy} = \eta \frac{dv}{dy} \tag{2.1}$$

Where η is a characteristic coefficient of the fluid and designate by dynamic viscosity [Pa.s].

Such hypothesis is only verified for Newtonians fluids and can be proven in experimental tests. When the linear proportionality is not verified due to the presence of macromolecules or severe conditions, the fluid is non-Newtonian. The relationship between strain rate and the shear stress is given by a rheological law. This characterizes the lubricant behaviour. For Newtonian lubricants the rheological behaviour is characterized by the Newton rheological law (2.1). The rheological behaviour of the non-Newtonian lubricants is not simple to characterize because the rheological model will be adapted to the flow type and the lubricant used.

2.4.1.1 *Thermoviscosity*

In general, the viscosity is strongly dependent on temperature. Water have slows viscosity variations in the order of 2.5% by °C and mineral oil about 10 to 20% by °C. Commonly, the oil viscosity has a large decrease when temperature increases.

Thermoviscosity is a characteristic of the oil and represents how the viscosity decreases with temperature. The equation that presents such variation is the ***ASTM D341*** standard, as shown in (2.2).

$$LogLog(\nu + a) = n - mLog(T) \tag{2.2}$$

Where v is the Kinematic viscosity (cst), T is the Temperature (Kelvin) and m, n, a are the constants dependent for each lubricant

The values of each constant are dependent on the lubricant and can be calculated as follows:

a=0.7 (cst), in case of mineral oil;

For calculate **m** and **n** handles to expression (2.2), we obtain:

$$m = \frac{\mathrm{Log}\left[\frac{Log(\nu_0 + a)}{Log(\nu_1 + a)}\right]}{Log\left(\frac{(\theta_1 + 273)}{(\theta_0 + 273)}\right)} \tag{2.3}$$

$$n = LogLog(\nu + a) + mLog(T) \tag{2.4}$$

2.4.1.2 *Viscosity Index*

The Viscosity Index specifies the reaction that the different classes of oils present with the variation of temperature. To obtain the Viscosity Index of an oil it is necessary to know the kinematic viscosity at two different temperatures.

$$V.I = \frac{L - U}{L - H} \times 100 \tag{2.5}$$

Where L is the kinematic viscosity at 40°C of a reference oil whose viscosity is vey dependent on temperature (V.I=100); H is the kinematic viscosity at 40°C of a reference oil whose viscosity has a small variation with the temperature (V.I=0) and U is the viscosity at 40°C of the oil that we intend to know the Viscosity Index.

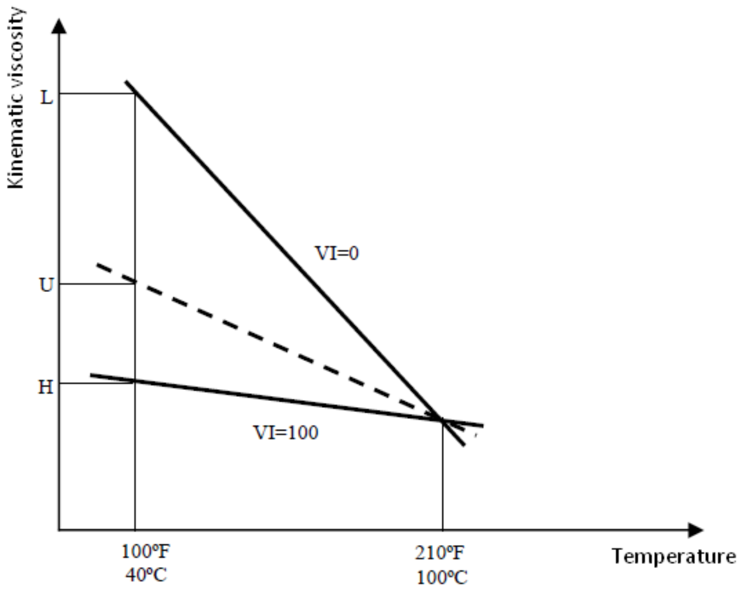

Figure 2: Viscosity index

2.4.1.3 *Piezoviscosity*

As with the temperature, the oil viscosity varies with pressure. The viscosity increases exponentially with the increase of pressure, being very dependent on the base oil used.

This value is very important for understanding the oil behavior under pressures found in EHD lubrication, as the film formation is attached with the piesoviscous properties.

A relation between pressure and viscosity can be calculated by ***Barus*** [1]law:

$$\eta_s = \eta_0 e^{\alpha p} \tag{2.6}$$

Where η_s is the Dynamic viscosity (Pa.s) at pressure p, η_0 is the Dynamic viscosity (Pa.s) at pressure p=0 and α is the Piezoviscosity coefficient (Pa^{-1}).

The piezoviscosity can be related with the kinematic viscosity through the empiric Gold´s [1] equation:

$$\alpha = s \times v^t \times 10^{-9} \tag{2.7}$$

Where v is the Kinematic viscosity (cst) at the operating temperature, s lubricant related constant ant t is the lubricant related constant.

Table 1 displays the lubricant dependent constants **s** and **t**.

Table 1: Lubricant dependent constants [1].

Constant	Mineral	PAO	PAG	Ester
s	9.904	7.3820	5.4890	6.6050
t	0.1390	0.1335	0.1485	0.1360

2.4.2 Other physical properties

2.4.2.1 *Bulk density or specific gravity*

The bulk density represents the ratio between mass of a body and its volume. The lubricant bulk density has an insignificant variation with the temperature but with the pressure is significant and very important to calculate the film thickness. The density or specific gravity is numerically equal to the bulk density and represents the ratio between lubricant and water bulk densities.

2.4.2.2 *Thermal conductivity*

The thermal conductivity quantifies the heat transmitted, due to a temperature gradient, through a unit thickness in a direction normal to a surface of unit area. This property varies linearly with the temperature.

2.4.2.3 *Specific heat*

The specific heat represents the amount of heat per unit mass required to raise the temperature by one Kelvin degree. A linear variation with temperature is verified. Such thermal conductivity is more significant with the increase of the polarity and hydrogen bonds to the molecules.

2.4.2.4 *Thermal capacity*

The thermal capacity supplied to an oil of unitary mass to increase the temperature of one degree ***Kelvin***.

2.4.2.5 *Thermal diffusivity*

The thermal diffusivity describes the heat propagation in oils. This value is obtained by ratio between thermal conductivity and the product of bulk density by specific heat.

2.4.2.6 *Freezing point*

The freezing point corresponds to the lowest temperature that the lubricant flows when is cooled. For mineral oils, the increase of the viscosity leads to increase the freezing point.

2.4.2.7 *Flashpoint*

Flashpoint is the minimum temperature that generates inflammation of the gases released, when nearing a naked flame.

2.4.2.8 *Combustion point*

Combustion point indicates the temperature that occur a permanent combustion due to the quantity of gases on oil surface.

2.4.2.9 *Auto-ignition point*

Auto-ignition point corresponds to the temperature which spontaneous combustion occurs (only in the presence of air).

2.4.2.10 *Drop point*

The drop point is a characteristic of grease and represents the temperature that occur a semi-solid state to liquid state transition.

2.5 Lubricating oils specifications

There are two specifications for lubricating oils:

- Viscosity specification;
- Service specification.

2.5.1 Viscosity specification

This specification can have two proposes and they are only based in oil viscosity:

1) Identification – they are specifications of refining or manufacturing taking in to account viscosity tolerances for certain ranges of viscosity;
2) Usage – They are imposed by costumers depending on the aplication. They are made by viscosity ranges at certain temperatures.

There are various professional societies that classify the lubricants by viscosity ranges, being the most common:

- ***SAE*** – Society of Automotive Engineers;
- ***ISO*** – International Standards Organization;
- ***AGMA*** – American Gear Manufactures Association;
- ***ASTM –*** American Society for Testing and Materials.

2.5.1.1 *SAE Viscosity classification*

The SAE classification is used for mineral oils (within or without additives) for motors, gearboxes and differentials. This is only classified according viscosity at particular temperature, not thus evaluating the oil quality.

2.5.1.2 *ISO Viscosity classification*

The *ISO* classification is very simple, because each class of viscosity is defined by a whole number. This whole number corresponds to kinematic viscosity of lubricant at 40 °C in centistokes (mm^2/s).

2.5.2 Service specifications

These classifications are described in function of a specific oil application. The more common usage classifications for lubricants are:

- ***API –*** American Petroleum;
- ***ACEA –*** *Association des Constructeurs Européens d'Automobiles*.

2.5.3 Environmental specifications

Due to high quantity of oil get into environmental without control there was a need to impose environmental specifications in order to render oils "environmental friendly".

With increase of environmental worries, comes the need to create certification systems with the intention of stimulate and promote concepts of recycling, biodegradability, resource preservation and pollution reduces is becoming more important.

Some examples of oil environmental certificates are:

- ***GreenMarck –*** Chinese certificate (Figure 3);
- ***Blauer Engel –*** German certificate (Figure 4).

Figure 3: *GreenMarck* symbol.

Figure 4: *Blauer Engel* symbol.

3 FZG test rig, gears, test procedure and wind turbine gear oils

3.1 FZG test rig

The experimental tests were performed in a FZG machine *(Forschugstelle für Zahnräder und Getriebebau*, English translation: research test rig for gears and transmissions). The FZG test rig uses a recirculating power loop principle, allowing a fixed torque to be applied to a pair of precision test gears [2].

Figure 5 displays a schematic representation of the test rig. The FZG test rig is composed by two gearboxes, test and drive gearbox (3), load clutch (4), two shafts, a driving motor and a torque measuring clutch (7). The pinion (1) and wheel (2) tests (situated in test gearbox) are linked to driving gearbox via two shafts. The load clutch, located in the middle of the front shaft, is used to apply torque with the aid of the load lever (6) and calibrated weights. The procedure is simple, it consists in the fixation of half of the load clutch, through the pin locker (5) and in the other half, it is applied the torque with the load lever and calibrated weights (depending on the desired load stage). The load lever is divided in seven positions which are 0.3, 0.35, 0.4, 0.45, 0.5, 0.55, and 0.6 m [3]. After applying the torque, the clutch screws are tightens and the load lever removed. The applied torque is measured through the torque measuring clutch (7), located on the other shaft. The DC motor can reach a maximum speed of 3000 rpm. The experimental test can be realized in dip or jet lubrication. When dip lubrication is used, the oil temperature can be controlled by electrical heaters mounted in test gearbox and the cooling coil and measured by the thermocouple (8). Under jet-lubrication, the oil flow is controlled by an external lubrication system.

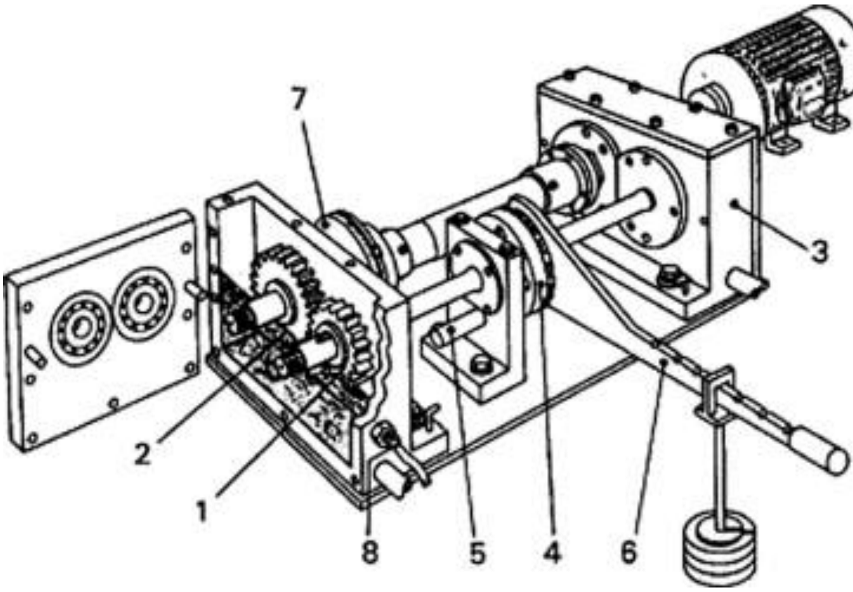

Figure 5: FZG test rig.

3.2 Gear Geometries

For the power loss tests helical gears were used with the designation “low loss gear” (see Figure 6 to Figure 8). These gears are designed in order to minimize the power loss using the following guidelines: (i) reduce the gear modules and increase the number of teeth, keeping constant the center distance using positive profile shifts, (ii) use standard cutting tools with 20° pressure angles, (iii) impose a total contact ratio close to 2.0, (iv) keep constant the gear safety factors against tooth root breakage (SF) and surface pressure (SH) [4].

In the drive gearbox type-C FZG spur gears with tooth width of 40 mm are used (see Figure 6). The gears parameters are exhibit in Table 2.

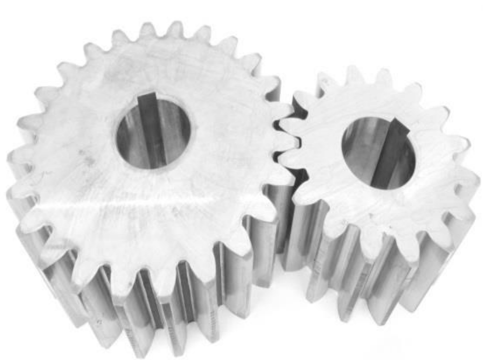

Figure 6: C40 gear design.

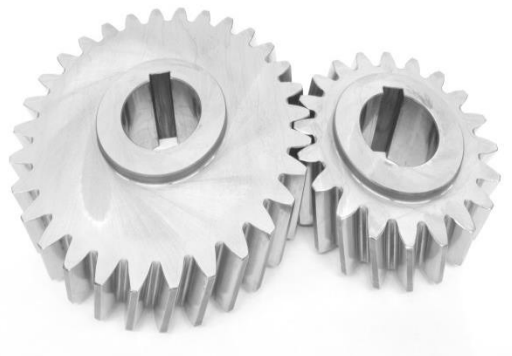

Figure 7: 501 gear design.

Figure 8: 951 gear design.

Table 2: Geometrical properties of C40, 501 and 951 gears.

Parameter	Symbol	C40		501		951	
		Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Number of teeth	z_1:z_2	16	24	20	30	38	57
Module [mm]	m	4.5		3.5		1.75	
Pressure angle [°]	α	20		20		20	
Helix angle [°]	β	0		15		15	
center distance [mm]	a	91.5		91.5		91.5	
Face width [mm]	b	40		23		23	
Addendum modification coef.	x_1:x_2	0.182	0.172	0.181	0.089	1.692	2.00
Addendum diameter [mm]	d_a	82.46	118.37	80.37	116.56	76.23	111.73
Transverse contact ratio [\]	ε_α	1.44		1.46		0.93	
Overlap ratio [\]	ε_β	0		0.54		1.08	
Total contact ratio [\ ]	ε_γ	1.44		2.00		2.02	
Material	\	20MnCr5		20MnCr5		20MnCr5	

3.3 Rolling Bearings and seals

The FZG machine gearboxes have rolling bearings and seals. The rolling bearings mainly ensure the positioning and load support of rotating parts, while the seals avoid oil leakage. The FZG gearboxes have the following rolling bearings:

- 2 Four point contact ball bearings QJ 308 N2MA;
- 6 Single row cylindrical roller bearings NJ 406.

For the tests performed with C40 gears, only cylindrical roller bearings were used, because there was no axial load.

For the low loss gear tests, the driving gearbox has four cylindrical roller bearings and the test gearbox has two cylindrical roller bearings and two four point contact ball bearings. The four point ball bearings absorb the axial load generated by the helical gears. The bearings properties are shown in Table 3.

Rotary lip seals are used to prevent oil leakage on the rotating shafts.

Table 3: QJ 308 N2MA and NJ 406 MA bearings properties.

Bearing type	Reference	C [KN]	C0 [KN]	D [mm]	d [mm]	B [mm]	MAX speed [rpm]
Four point contact ball bearing	QJ 308 N2MA	78	64	90	40	23	14000
Single row cylindrical roller bearing	NJ 406 MA	34.1	34	90	30	23	11000

The FZG machine has a total of five seals. The driving gearbox has three seals, being the seal on the motor side different from others (26 mm shafts diameter), and the test gearbox has two seals. All seals are made with the same material (Viton lip seals) differing only in inner diameter. The characteristics of the seals are presented in Table 4.

Table 4: Gearboxes seals characteristics.

seal type	Reference	Lip Material	d [mm]	D [mm]	b [mm]	max speed [rpm]
Radial shaft seals	30X47X7 HMS5 V	V	30	47	7	8913
Radial shaft seals	26X47X7 HMSA10 V	V	26	47	7	7346

3.4 Measurement equipment

During the power loss tests the torque loss and the operating temperatures were measured. For measure and record the operating temperatures at specific points of the assembly, type K thermocouples were used, as shown on Figure 9. The temperatures acquisition was made after data acquisition board with eight devices and the temperature that each device measures is presented in Table 5.

The torque loss was measured using an ETH Messtechnik DRDL-II torque transducer assembled on the FZG test rig between the drive gearbox and the DC motor. To acquire data the system uses a sensor interface (*ValueMaster V. 2.43*), communicating with a PC via Ethernet. The technical characteristic of the sensor are exhibited on Table 6.

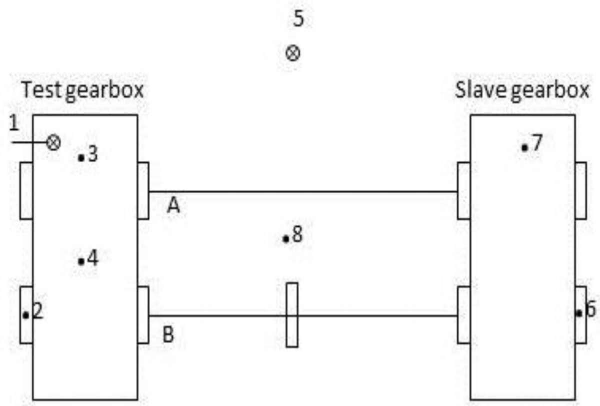

Figure 9: FZG with thermocouples.

Table 5: Temperature nomenclature.

Number	Device	Temperature measured
1	Dev_1_ai0	Out Oil
2	Dev_1_ai1	Test gearbox bearing case
3	Dev_1_ai2	Test gearbox cover
4	Dev_1_ai3	Oil injection
5	Dev_2_ai0	Room
6	Dev_1_ai1	Drive gearbox bearing case
7	Dev_1_ai2	Drive gearbox cover
8	Dev_1_ai3	Mass temperature

Table 6: Technical specifications of the ETH DRDL torque cell.

Torque transducer Type DRDL-II	
Nominal torque [N.m]	50
Measurement range [N.m]	10/50
Non-linearity [%]	<0,1
Hysteresis [%]	<0,1
Accuracy [%]	0,01
Temperature sensitivity [%/K]	0,01
Torque Measuring Module Type ValueMasterBase	
Accuracy [%]	0,02
Non-linearity [%]	0,1
AD converter resolution	11 bit + 1 bit for leading sign

3.5 Operating conditions

The operating conditions consisted in five FZG standard load stages (see Table 7 [5]). These are applied in load clutch with the aid of the load lever arm (lever arm 0.35 m) and calibrated weights (see Table 9 [3]). For each load stage the test run at 200 rpm, 400 rpm and 1200 rpm which corresponds to tangential speed of 1.14 m/s, 2.28 m/s and 6.83 m/s. All of these operating conditions are displayed in Table 7.

Table 7: FZG load stages used on the torque loss tests.

Load stage FZG	torque on pinion (T1) [N.m]	torque on wheel (T2) [N.m]
k1	3.30	4.95
k5	69.98	104.98
k7	132.48	198.72
k9	215.57	323.35
k11	319.26	478.89

The torque value, for each load stage, can be calculated using equation 3.1 [3].

$$T_1 = T_H + (F_K + F_W) \times h \tag{3.1}$$

Where:

- T_1 [Nm] – Torque on pinion;
- T_H [Nm] – Torque of load lever, two levers are available with 3.3 Nm (K_1) and 13.7 (K_2);
- F_K [N] – weight of the load support rod which is 43.1 N for the available rod;
- F_W [N] – weight of the loading weights see Table 9;
- h [m] – effort arm for F_k and F_w, with grooves in the load lever;

For load stage K_3 the heavier load lever is used and the load support rod placed in the load lever on the position 0.35 m.

$$T_1 = 13.7 + (43.1) \times 0.35 = 28.79\ Nm \tag{3.2}$$

For load stage K_5 it is necessary two weights, the load lever, the load support rod on the position 0.35 m.

$$T_1 = 13.7 + (43.1 + 51 + 66.7) \times 0.35 = 69.98\ Nm \tag{3.3}$$

Table 8: Operating conditions on the torque loss tests.

load stage FZG	T_2 [N.m]	n_2 [rpm]	v_t [m/s]	P_{in} [W]
k1	4.95	200	1.14	103.67
		400	2.28	207.35
		1200	6.83	622.04
K5	104.97	200	1.14	2198.49
		400	2.28	4396.97
		1200	6.83	13190.92
K7	198.68	200	1.14	4161.20
		400	2.28	8322.39
		1200	6.83	24967.18
K9	323.27	200	1.14	6770.45
		400	2.28	13540.89
		1200	6.83	40622.68
K11	478.89	200	1.14	10029.85
		400	2.28	20059.70
		1200	6.83	60179.09

Table 9: Calibrated weights used for composing the FZG load stages.

No	Weights	
	[Kgf]	[N]
1	5.2	51.0
2	6.8	66.7
3	8.4	82.4
4	9.8	96.1
5	11.4	111.8
6	12.8	125.6
7	14.4	141.3
8	15.8	155.0
9	17.2	168.7

3.6 Wind turbine gear oils

For this work eight wind turbine gear oils were tested, of which 4 are Polialphaolefin base oils (PAOX, PAOR, PAOC and PAOM), two Mineral base oils (MINR, MINE), one Polyalkyleneglycol base oil (PAGD) and one esters base oil (ESTR) , presenting all a viscosity grade ISO VG 320. The properties of each lubricant are provided by manufacturers, but to assure a better comparison between them, the lubricants were submitted to density and viscosity measurements. The physical properties of lubricants used are shown in Table 10.

Looking to Table 10, we can see the differences between the oils with the same base oil (PAOR, PAOX, PAOM and PAOC). As said previously this oils have the same viscosity grade but the measurements show a significant difference between them (PAGD – 289.13 cst; PAOM – 325.96 cst). Otherwise the values measured are within the standard deviations for the viscosity grade announced.

Table 10: Physical properties of the tested oils.

Parameter	Units	PAOR	PAOX	PAOC	PAOM	MINR	MINE	ESTR	PAGD
Density (15°C)	*g/cm³*	0.860	0.855	0.861	0.863	0.902	0.893	0.914	1.059
Viscosity (40°C)	*cSt*	313.52	297.20	304.09	325.96	319.22	328.30	301.93	289.13
Viscosity (80°C)	*cSt*	60.41	62.62	59.33	64.63	43.94	66.48	56.13	78.89
Viscosity (100°C)	*cSt*	33.33	35.42	32.86	35.89	22.33	37.13	30.71	48.09
Viscosity index	-	150	167	151	158	85	163	140	230
Thermoviscosity (40°C)	K^{-1}	0.0513	0.0481	0.0509	0.0502	0.0641	0.0494	0.0527	0.0391
Thermoviscosity (80°C)	K^{-1}	0.0328	0.0313	0.0326	0.0324	0.0380	0.0321	0.0334	0.0269
Thermoviscosity (100°C)	K^{-1}	0.0269	0.0259	0.0267	0.0267	0.0301	0.0254	0.0272	0.0227
Piezoviscosity (40°C)	Pa^{-1}	1.59E-8	1.58E-8	1.58E-8	1.60E-8	2.21E-8	1.60E-8	1.44E-8	1.27E-8
Piezoviscosity (80°C)	Pa^{-1}	1.28E-8	1.28E-8	1.27E-8	1.29E-8	1.68E-8	1.29E-8	1.14E-8	1.05E-8
Piezoviscosity (100°C)	Pa^{-1}	1.18E-8	1.19E-8	1.18E-8	1.19E-8	1.53E-8	1.20E-8	1.05E-8	0.98E-8
Thermal expansion coefficient	-	-5.5E-4	-7.5E-4	-7.4E-4	-7.0E-4	-5.8E-4	-7.0E-4	-8.0E-4	-7.1E-4

Figure 10 shows the variation of the viscosity with the temperature. For oils with the same base oil doesn't no significant differences are observed. For high temperature PAGD present a higher viscosity than the others, the MINR present an opposite behaviour. The other oils present intermediate values.

Looking now to Figure 11, it is possible see that the PAGD oil exhibits a highest density variation with the temperature variation despite having the highest density and has the highest density then other.

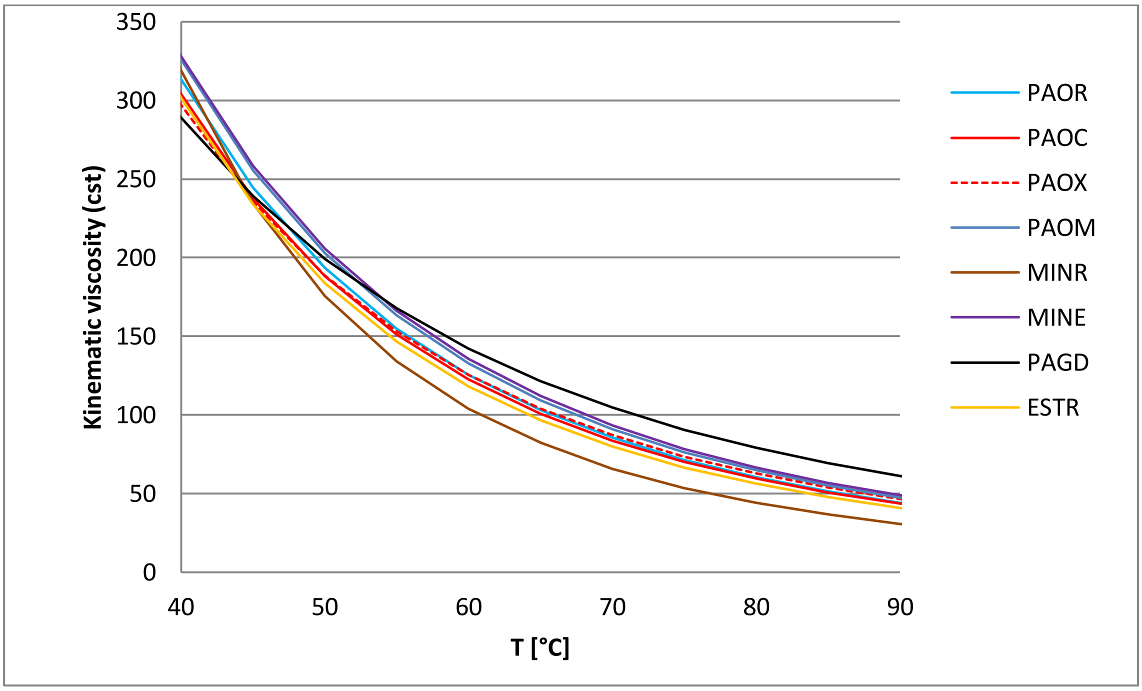

Figure 10: Variation of the kinematic viscosity with temperature.

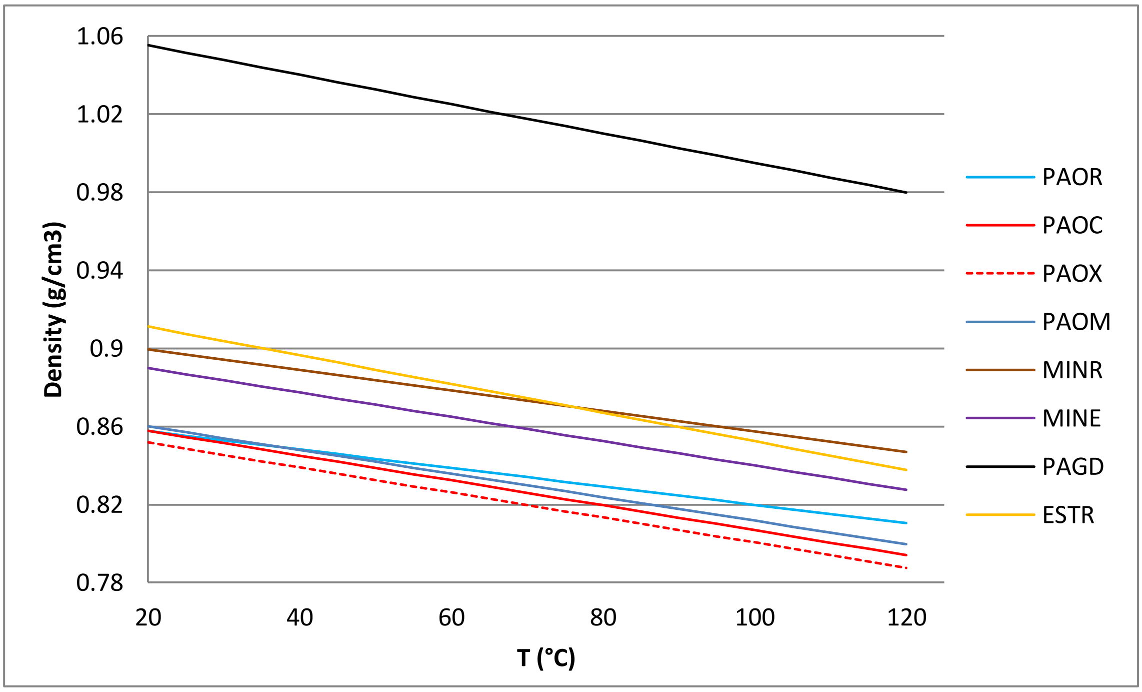

Figure 11: Variation of the density with the temperature.

4 Experimental tests

4.1 Introduction

This section describes the experimental tests performed on the FZG test rig, with different oil formulations and different gear geometries, and presents the results obtained.

The energetic efficiency and the torque loss will be the main focus on the results obtained for each combination load / speed / gear geometry / oil formulation tested.

4.2 Test procedure

The test procedure for the gear torque loss test can be expressed in the following steps:

1. Application of the load stage (K1, K5, K7, K9, K11);
2. Set speed value (200, 400 and 1200rpm)
3. The duration of each Load-speed combination was 3 hours, while the torque loss and the assembly operating temperatures were continuously measured and recorded.

An the end of each load-speed stage, the average values of the last half hour of test were calculated to obtain the torque loss and operating temperatures for that stage. The other values were not considered because only in the last half hour all temperatures were stabilized, i.e. the test rig was operating in steady state conditions.

During the power loss tests the jet oil temperature was monitored and was kept at 80 °C with a volume flow of 3 l/min. To assure a faster stabilization temperature from one day to the next, the oil circulation system was operating continuously during the night at constant temperature. The test room has a ventilation system to ensure the air recirculation and thus, contributing to maintain a stabilized environment temperature.

At the end of each test, the test gearboxes and its components were thoroughly cleaned. The oil reservoir was drained and together with the other components, cleaned with solvent.

In order to ensure that the gearbox was thoroughly cleaned, it was filled with solvent and thereafter drained by the return system. Finally, the first 3 litters of new oil passing the oil jet circulation system were collected to ensure the removal of all used oil and solvent.

The block diagram in Figure 12 resumes the test procedure.

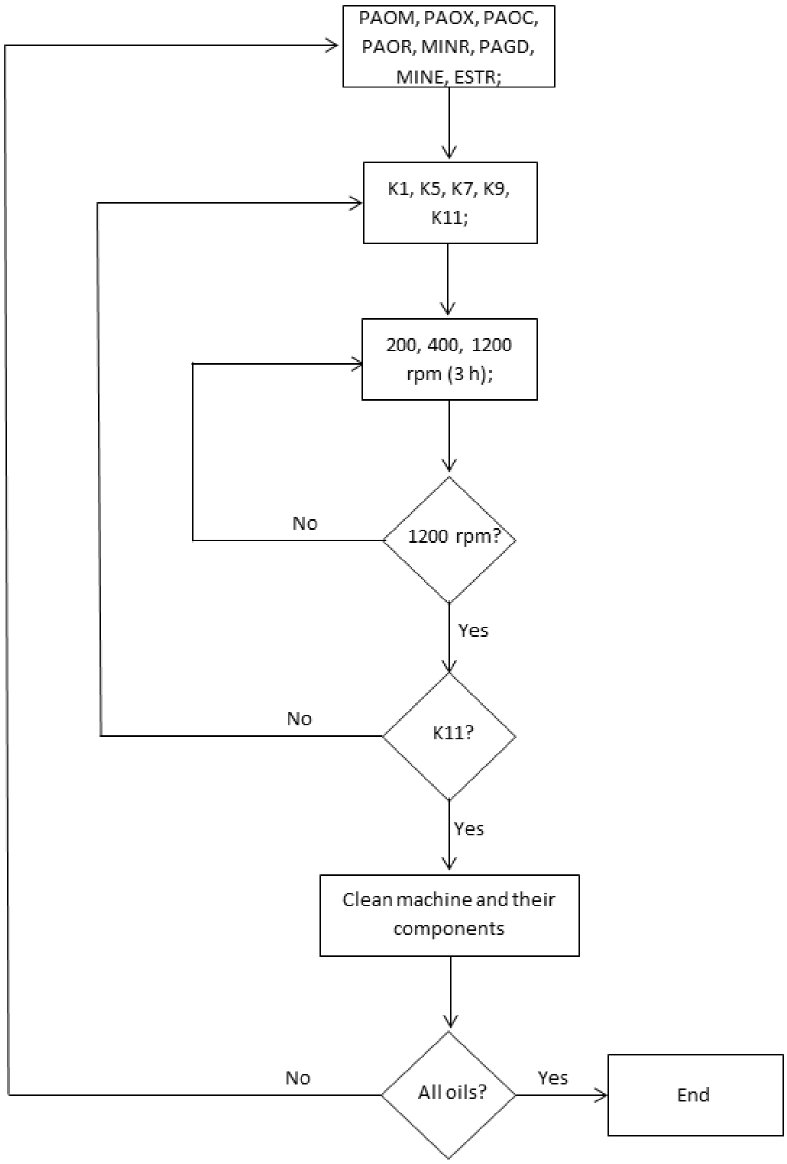

Figure 12: Schematic test sequence.

4.3 Performance Evaluation of the wind turbine oils

This section displays the experimental results obtained and the comparison of the oils tested. Each oil type was tested in FZG test rig with the gear geometries C40, 501 and 951. The power loss tests were performed with oil injection at a constant temperature of 80°C, therefore all oil properties dependent of the oil temperature are constant during all tests.

The torque loss measured includes all existent losses on the FZG test rig, essentially the sum of the torque loss of the test gearbox and the slave gearbox. In this work the gears with the different geometries were only installed on the test gearbox, i.e. the slave gearbox always used C40 gears.

So, in a first phase it was necessary to test all the oils with C40 gears installed on both the test and the slave gearbox with all oils. This procedure allows knowing the torque loss on the slave gearbox, which is half of the total torque loss measured. In a second phase the helical gears were installed in the test gearbox and subtracting the C40 slave gearbox torque loss (first phase) from the total torque loss measured in the second phase, the helical gearbox torque loss is known.

The experimental tests performed to evaluate the power loss in the slave gearbox with the gear geometry C40 (test and slave gearbox using C40 gears) were performed in a previous work. All the tests with helical gears were performed by the author.

4.3.1 C40 gear design

Table 11 displays the torque loss for the test gearbox (half of the total torque loss measured) for each load-speed stage and Table 12 displays the efficiency of the test gearbox. The procedure to calculate the efficiency is detailed in A.1.

Table 11: Torque loss on the test gearbox with C40 gears [N.m].

Speed [rpm]	Load stage	PAOR	PAOC	PAOM	PAOX	MINR	MINE	PAGD	ESTR
200	k1	0.61	0.64	0.81	0.72	0.58	0.62	0.69	0.62
	k5	1.56	1.49	1.73	1.60	1.86	1.48	1.54	1.57
	k7	2.54	2.47	2.69	2.60	2.96	2.66	2.35	2.43
	k9	3.86	3.89	4.08	3.71	4.44	4.04	3.50	3.78
400	k1	0.69	0.75	0.94	0.90	0.75	0.78	0.87	0.76
	k5	1.61	1.56	1.80	1.69	1.88	1.57	1.66	1.65
	k7	2.48	2.51	2.66	2.51	2.87	2.64	2.39	2.42
	k9	3.60	3.74	3.87	3.68	4.27	3.96	3.33	3.67
1200	k1	1.06	1.12	1.17	1.24	1.07	2.22	1.13	1.10
	k5	2.03	1.96	2.09	2.00	2.11	3.88	2.21	2.05
	k7	2.78	2.77	2.90	2.72	2.93	5.35	2.88	2.68
	k9	3.67	3.75	3.85	3.70	4.04	7.84	3.59	3.66

Table 12: Test gearbox efficiency with C40 gears.

Speed [rpm]	Load stage	PAOR	PAOC	PAOM	PAOX	MINR	MINE	PAGD	ESTR
200	k1	0.869	0.862	0.820	0.843	0.875	0.865	0.849	0.866
	k5	0.985	0.986	0.983	0.985	0.982	0.986	0.985	0.985
	k7	0.987	0.987	0.986	0.987	0.985	0.987	0.988	0.988
	k9	0.988	0.988	0.987	0.988	0.986	0.987	0.989	0.988
400	k1	0.850	0.835	0.788	0.797	0.835	0.828	0.806	0.832
	k5	0.985	0.985	0.983	0.984	0.982	0.985	0.984	0.984
	k7	0.987	0.987	0.987	0.987	0.985	0.987	0.988	0.988
	k9	0.989	0.988	0.988	0.989	0.987	0.988	0.990	0.989
1200	k1	0.755	0.740	0.725	0.707	0.754	0.743	0.738	0.744
	k5	0.980	0.981	0.980	0.981	0.980	0.981	0.979	0.980
	k7	0.986	0.986	0.985	0.986	0.985	0.986	0.985	0.986
	k9	0.989	0.988	0.988	0.988	0.987	0.988	0.989	0.989

To allow better understanding of the performance of each oil type, a comparison between PAOs, i.e. oils with the same base oil, and a comparison between all the other oils, using PAOR as the reference of the PAO based oil, thus analysing the performance of different base oil.

4.3.1.1 *Comparison of PAO base oils*

The load stage K1 was performed in order to quantify the no-load losses for each rotational speed. Figure 13 represents the torque loss of C40 gears in load stage K1. The PAOR and PAOC present the lowest torque loss, independently of the speed. At a rotational speed of 200 rpm and 400 rpm, PAOM has the highest torque loss while at 1200 rpm the PAOX has the highest torque loss.

For load stage K5 (Figure 14), the lowest torque loss value was attained by PAOC and the highest by PAOM. For stage K7 (Figure 15), PAOM still displays the highest value of torque loss. The oil that has the lowest torque loss, changes with the input speed, was PAOR at 200 rpm, PAOX at 400 rpm and PAOC at 1200 rpm.

As already observed, PAOM have the highest value of torque loss for load stage K9 (Figure 16). For low input speed, PAOX distinguishes itself from others with lower torque loss. Between 400rpm and 1200 rpm the torque loss remained almost constant and for this range of input speed the PAOR had the lowest torque loss.

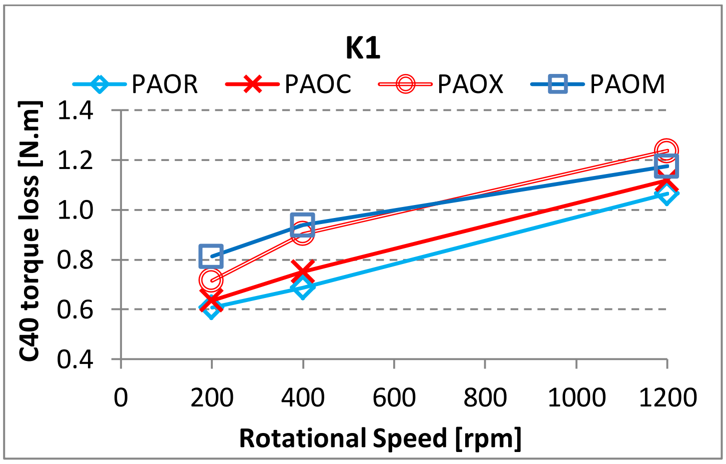

Figure 13: Torque loss of C40 gear at load stage K1 lubricated with PAO's.

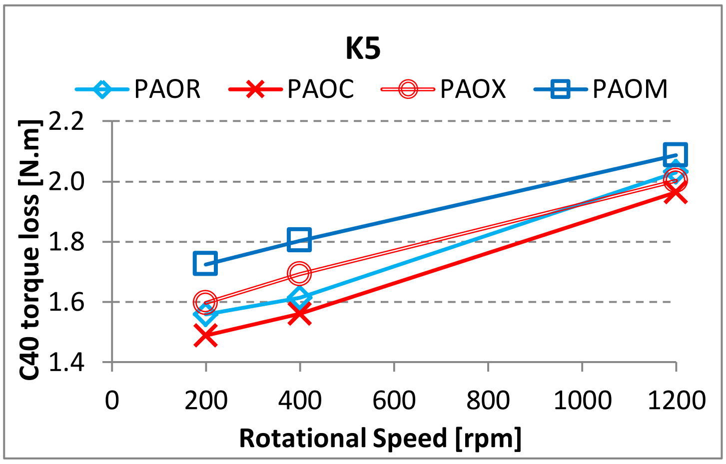

Figure 14: Torque loss of C40 gear at load stage K5 lubricated with PAO's.

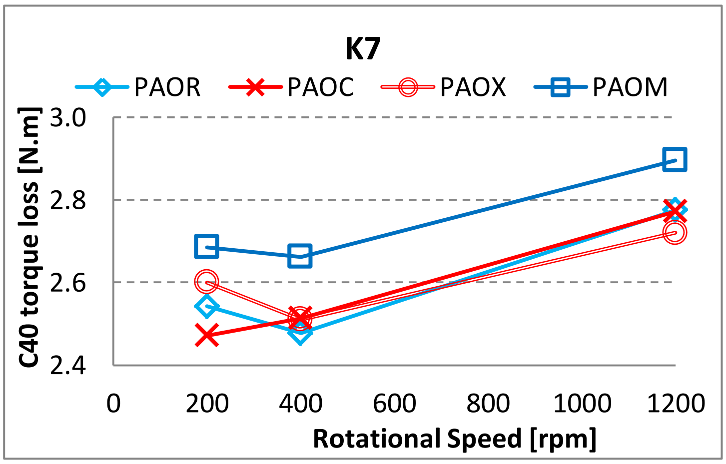

Figure 15: Torque loss of C40 gear at load stage K7 lubricated with PAO's.

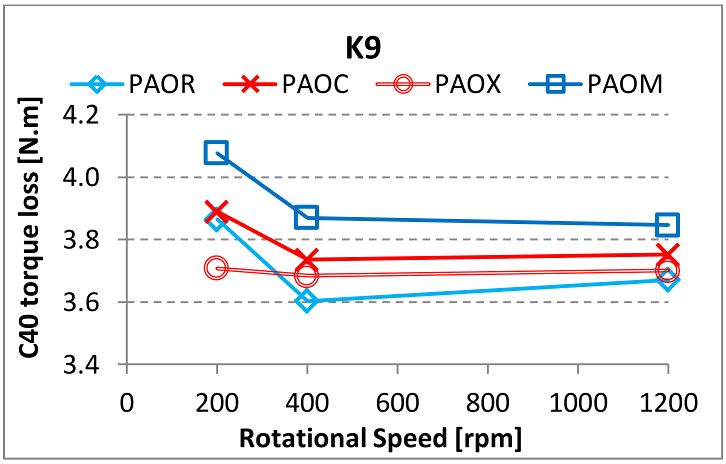

Figure 16: Torque loss of C40 gear at load stage K9 lubricated with PAO's.

The oil that presents the best efficiency at load stage K1 was PAOR (Figure 17), decreasing with the increasing speed. PAOM presents the lowest efficiency whatever the speed. The PAOC oil had efficiency close to PAOR.

For load stage k5 (Figure 18), the PAOC oil has the highest efficiency, independently of speed. As in load stage K1, PAOM displayed the lowest efficiency. The PAOR and the PAOX displayed approximately the same efficiency.

For load stage k7 (Figure 19), the PAOM remains the oil with the lowest efficiency. At 200 rpm, PAOX distinguished itself with high efficiency and, for remaining range speed, the PAOR, the PAOX and PAOC had approximately the same efficiency.

For high load, k9 load stage (Figure 20), the oil that displayed the lowest efficiency is PAOM and the one with higher efficiency was PAOX at 200 rpm and PAOR at the speed range from 400 to 1200 rpm. The maximum differences between the oils efficiencies were very small, around 0.05%.

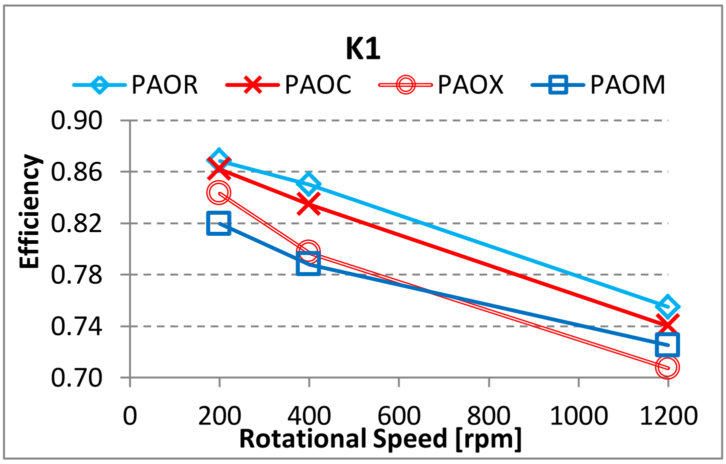

Figure 17: Efficiency of C40 gear at load stage K1 lubricated with PAO's.

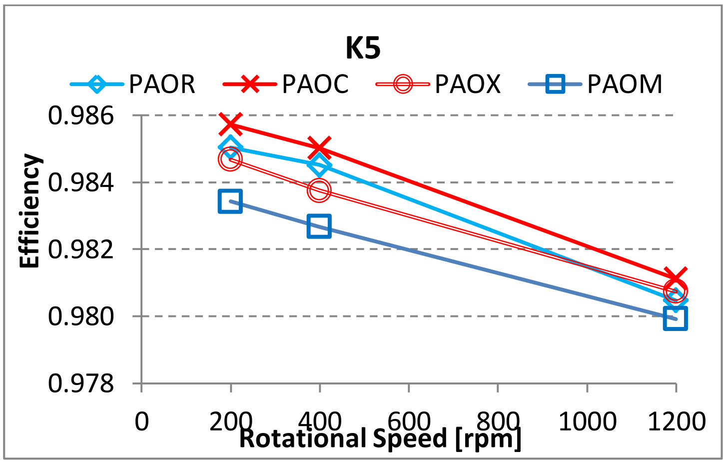

Figure 18: Efficiency of C40 gear at load stage K5 lubricated with PAO's.

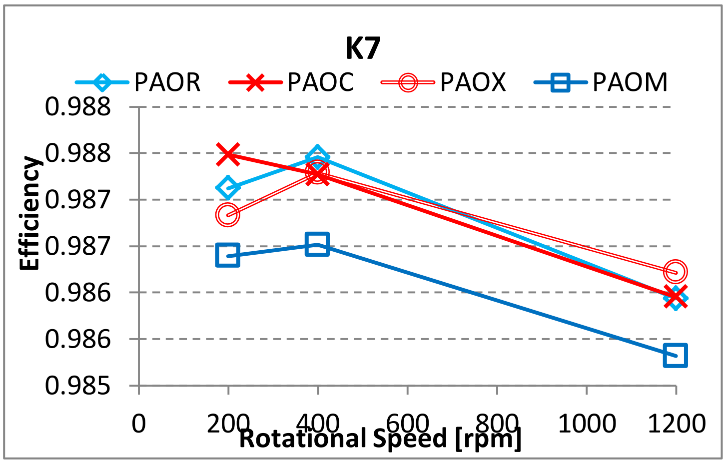

Figure 19: Efficiency of C40 gear at load stage K7 lubricated with PAO's.

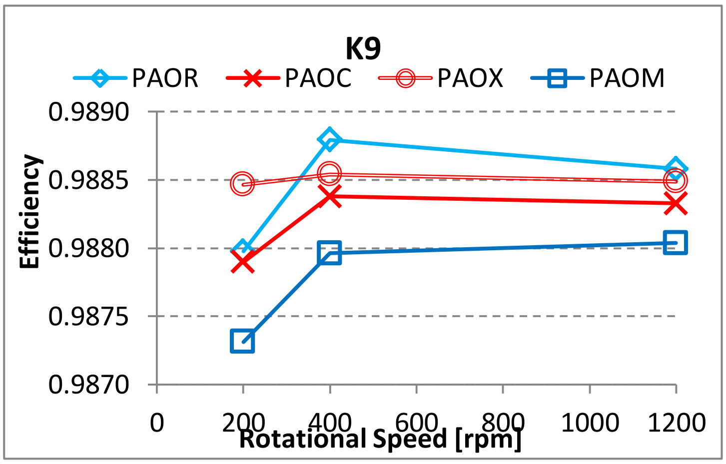

Figure 20: Efficiency of C40 gear at load stage K9 lubricated with PAO's.

4.3.1.2 *PAOR, MINR, MINE, PAGD and ESTR performance*

This section presents the experimental results the performance of each oil base, i.e. the performance of mineral oil base, ester oil base, PAO oil base and PAG oil base.

In load stage K1 (Figure 21), no-load losses representative test, the oil that displays the highest torque loss is the PAGD, independently of the speed. The oil that present the lowest Torque loss varies with speed. At 200 rpm the MINR present the lowest value, and at range speed 400 rpm to 1200 rpm PAOR that present the lowest torque loss.

In Figure 22, representation of the load stage k5, the lowest value of torque loss is given by MINE for all speed range. The worst oil is the MINR for range speed 200 to 400 rpm and PAGD for 1200 rpm.

For K7 load stage (Figure 23) the worst performance is present by MINR. PAGD, for the same speed range 200 to 400 rpm, and, despite the high viscosity, present the lowest torque loss. At 1200 rpm ESTR and MINR, with approximately torque loss, present the lowest torque loss.

For K9 load stage (Figure 24) PAGD have the lowest value of torque loss and MINR have the highest value of torque loss, i.e. PAGD is the best change and MINR is the worst change for high load stage.

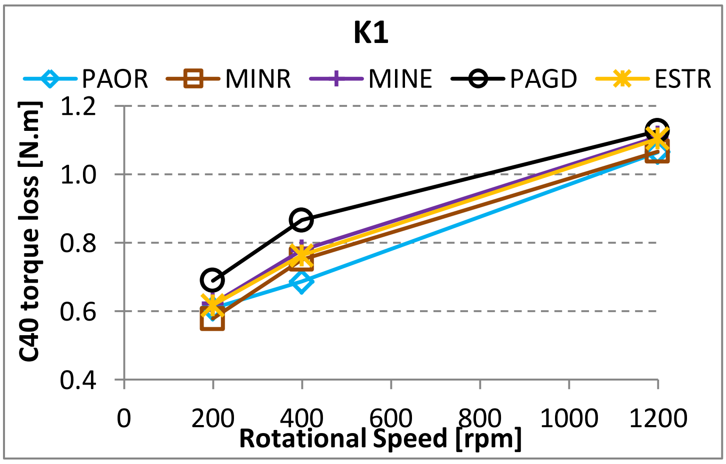

Figure 21: Torque loss of C40 gear at load stage K1.

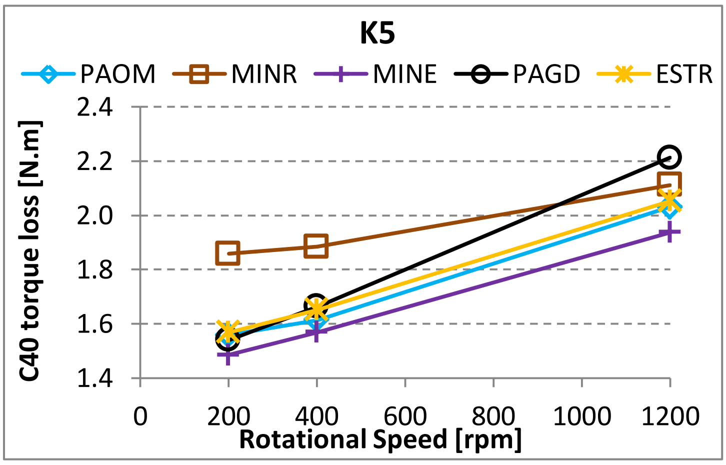

Figure 22: Torque loss of C40 gear at load stage K5.

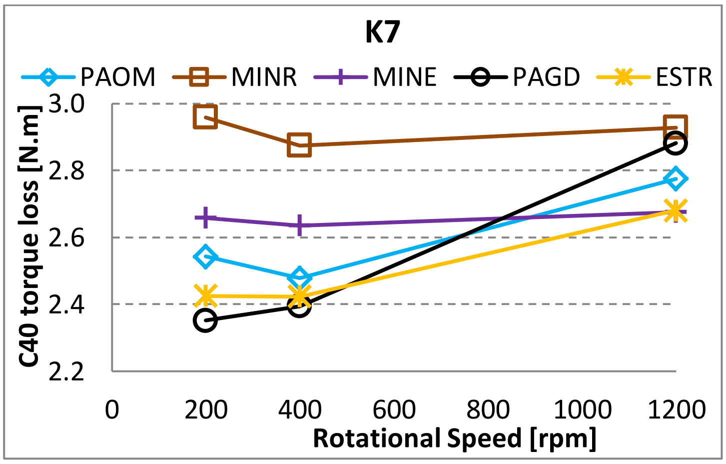

Figure 23: Torque loss of C40 gear at load stage K7.

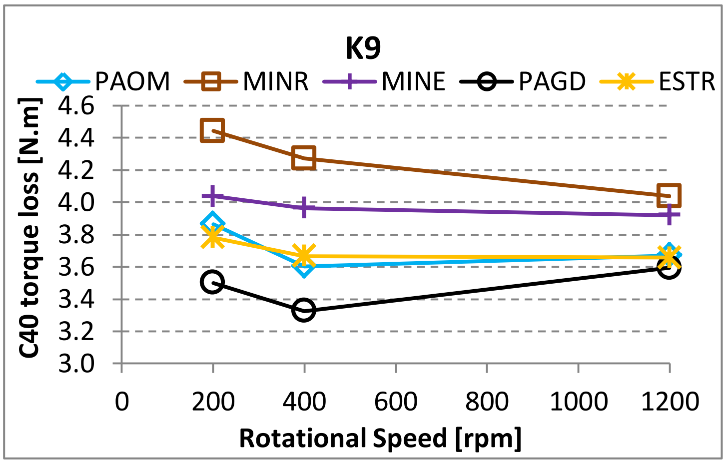

Figure 24: Torque loss of C40 gear at load stage K9.

In the no-load representative stage, k1 load stage (Figure 25), the oil that presents the best efficiency is PAOR, at speed range 400 to 1200 rpm, and the worst efficiency is PAGD, for all speed range. The MINR, at 200 rpm and 1200, also present a good efficiency being the best at 200 rpm.

With load increasing, k5 load stage (Figure 26), the MINE distinguished with the best performance for all speed range. At range speed 200 to 400, MINR is the worst oil with low efficiency and, at 1200 rpm, the low efficiency is presented by PAGD. The PAOR and the ESTR has, approximately, the same efficiency.

At range speed 200 to 400 in the load stage k7 (Figure 27), MINR and PAGD present the lowest and the highest efficiency, respectively. At 1200 rpm MINR remains with low efficiency and the high efficient is present by ESTR and MINE.

In load stage k9 (Figure 28), in spite of the small differences (0.3%), PAGD distinguished by high efficiency and MINR by low efficiency. As has already been seen previously the PAOR and ESTR has the same efficiency approximately.

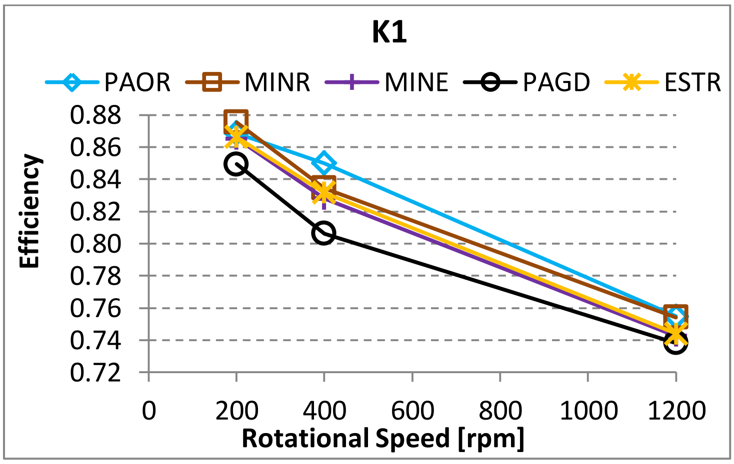

Figure 25: Efficiency of C40 gear at load stage K1.

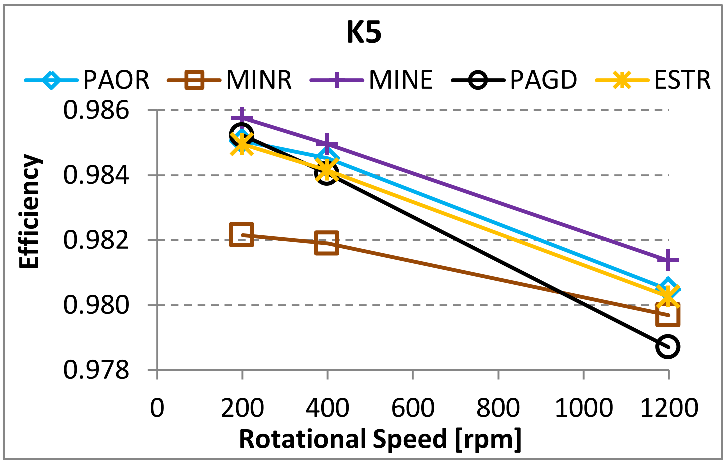

Figure 26: Efficiency of C40 gear at load stage K5.

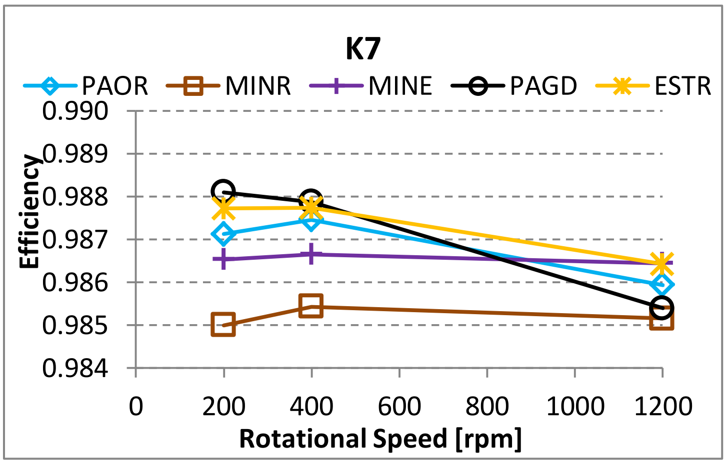

Figure 27 Efficiency of C40 gear at load stage K7.

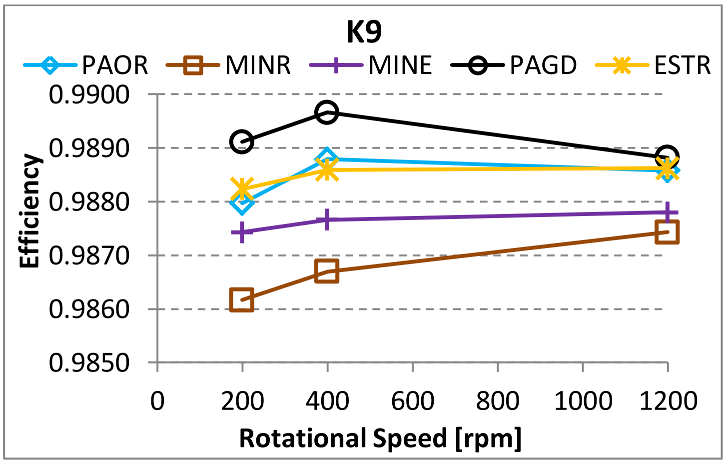

Figure 28: Efficiency of C40 gear at load stage K9.

4.3.1.3 *Results discussion*

In load stage k1 (Figure 13), the PAOR and PAOC present the lowest torque loss can be due the lowest viscosity at the injection temperature (80 °C). The performance of PAOM PAOX can be explained because have the highest viscosity). The highest torque loss of PAOX at 1200 can be associated to the change of the regime of lubrication.

In load stage K5 (Figure 14) is notorious the viscosity dependency for 200 rpm and 400 rpm (PAOC the lowest viscosity value and PAOM the highest viscosity value). At 1200 rpm this tendency isn't fully verified it can be due to the improvement of conditions be forming film.

For stage K7 (Figure 15) despite the small difference between viscosity oils at 200 rpm is in the same verified the viscosity dependency as was previously verified being eliminates with the increase of the speed (400 rpm and 1200 rpm), shown a dispositions according to film thickness.

The bed performance of PAOM and the good performance of PAOR in load stage k9 (Figure 16) can be due of the regime lubrication conditions.

In load stage K1 (Figure 21), no-load losses representative test, the bed performance of PAGD is due to the high viscosity. At 200 rpm the oils have a growing disposition according to viscosity (MINR the lowest viscosity and PAGD the highest viscosity). The PAOR good performance at range speed 400 rpm to 1200 rpm can be due to the approximation to the lubrication regime change. At 1200 rpm, as all oil had changed of lubrication regime, the oils disposition is again according to the viscosity.

With the MINE performance in load stage k5 (Figure 22), it can affirmed that for low load stage the MINE is the best oil. The oil disposition is completely different of the K1 stage. This new disposition may be due to poor conditions for film formation.

In K7 load stage (Figure 23) and for range speed 200 to 400 rpm the ESTR, PAOR and MINE present a viscosity dependency. The disposition of the oil for this speed range can be due to the poor conditions to form the film EHD. In 1200 rpm the viscosity dependency do not occur and the new disposition can be due the change of lubrication regime.

For K9 load stage (Figure 24) the PAOR, the ESTR and MINE are arranged according to their viscosity. The PAGD good performance and the bed performance of MINR are linked to the high film thickness.

4.3.1.4 *Film thickness calculation*

To calculate the centre film thickness and thus calculate the specific film thickness, Dowson and Higginson [1] equation for linear contact (equation 4.1) was used.

$$h_0 = 1.949 \frac{(\eta_{bulk}\alpha_{bulk}(U_1 + U_2))^{0.727} {R_X}^{0.364} (E^* l)^{0.091}}{{F_N}^{0.091}} \tag{4.1}$$

The centre film thickness was corrected using the thermal reduction factor (equation 4.2 to equation 4.5).

$$h_{0T} = \phi_T h_0 \tag{4.2}$$

$$\phi_T = \left(1 + 0.1\left(1 + 14.8{V_e}^{0.83}\right)L^{0.64}\right)^{-1} \tag{4.3}$$

$$V_e = \frac{|U_1 - U_2|}{|U_1 + U_2|} \tag{4.4}$$

$$L = \frac{\beta_{bulk}\eta_0(U_1 + U_2)^2}{K_f} \tag{4.5}$$

Where, β is the thermoviscosity coefficient, K_f is the thermal conductivity and η_0 is the dynamic viscosity at pressure p=0.

The specific film thickness represents the ratio between the theoretical centre film thickness and composite surface roughness (σ≈1 μm) (equation 4.6).

$$\Lambda = \frac{h_{0t}}{\sigma} \tag{4.6}$$

The oil temperature inside the contact, designated by bulk temperature, is calculated according DIN 3990. This temperature (Table 13) will be used to calculate the oil properties and thus calculate the specific film thickness.

$$T_{flash} = 0.08{T_1}^{1.2}\left(\frac{100}{\nu_{40}}\right)^{{\nu_{40}}^{-0.4}} \tag{4.7}$$

$$T_{bulk} = XS(T_{oil} + C_1 T_{flash}) \quad (4.8)$$

With:

- XS=1.2, for jet-lubrication
- XS=1, for dip lubrication
- C_1=0.7

Table 13 displays the bulk and flash temperature calculated for the PAO oils at all load stages

Table 13: Flash and bulk temperature on C40 for each test performed lubricated with PAO's.

	PAOR		PAOX		PAOC		PAOM	
Load stage	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}
K1	96.65	0.30	94.96	0.30	96.12	0.30	96.29	0.30
K5	105.70	11.68	105.37	11.71	105.73	11.70	105.55	11.66
K7	117.17	25.11	115.70	25.19	116.93	25.16	116.96	25.06
K9	133.74	45.04	133.89	45.18	134.21	45.12	134.12	44.95

Figure 29 show the specific film thickness for each test conditions. The specific film thickness, for each oil type, is similar because they have the same index viscosity. The small differences are due to the different additive package.

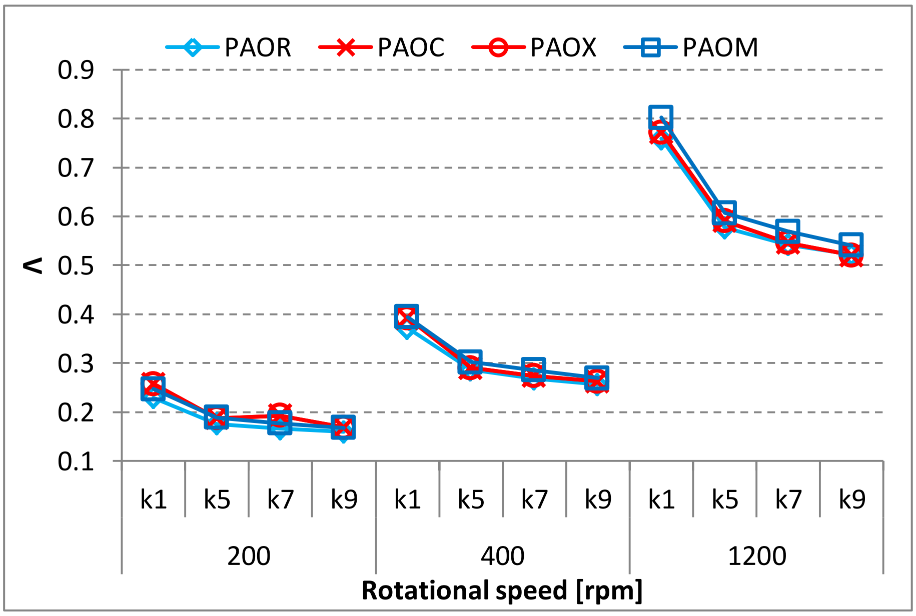

Figure 29: Specific film thickness for C40 gears lubricated with PAO's.

Table 14 displays the bulk and flash temperature calculated for the PAO oils at different load stages.

Table 14: Flash and bulk temperature on C40 for each test performed.

	PAOR		MINE		MINR		PAGD		ESTR	
Load stage	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}
K1	96.65	0.30	96.46	0.30	96.90	0.30	96.68	0.30	96.29	0.30
K5	105.70	11.68	105.72	11.65	106.11	11.67	106.00	11.73	105.65	11.70
K7	117.17	25.11	117.10	25.05	117.17	25.10	117.20	25.23	117.27	25.17
K9	133.74	45.04	133.73	44.93	133.89	45.01	134.19	45.25	134.14	45.14

Regarding the specific film thickness (Figure 30) the MINR, MIRN and ESTR have a relative behaviour according to their viscosity. The PAGD has the highest specific film thickness, even though having the lowest piezoviscosity. The MINE has the second highest specific film thickness due its high viscosity and high piezoviscosity.

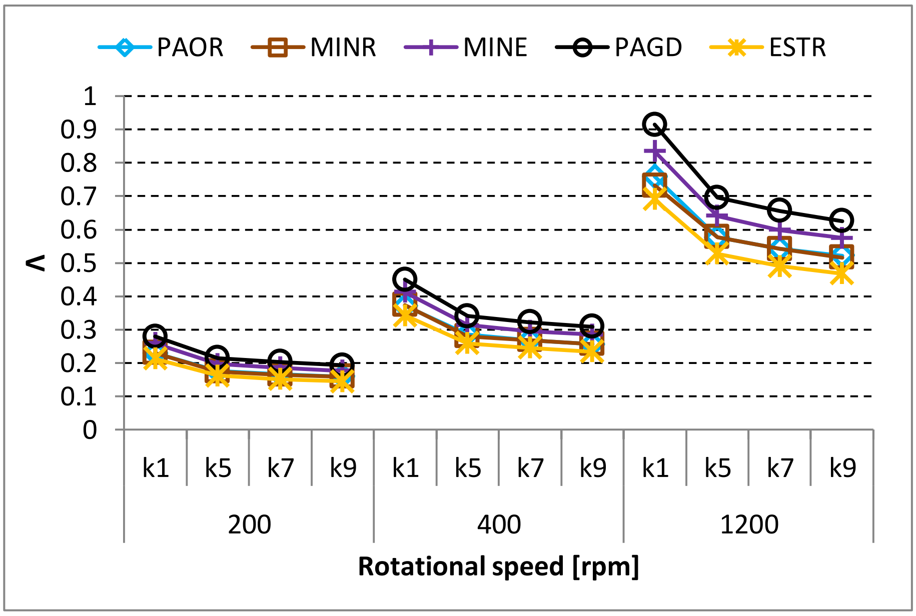

Figure 30: Specific film thickness for C40 gears.

4.3.2 501 gear design

The results will be presented in two groups, one with PAO's and other with all base oils, just as presented for C40 gears.

The torque loss, for the different oils tested, is presented in Table 15 and the test gearbox efficiency is present in Table 16. The torque loss presented corresponds to the total torque loss measured (test gearbox with 501 gear and slave gearbox with C40 gears) subtracted by half of the torque loss measured with C40 gears (presented in previous section), ie the torque loss represents only the losses on the test gearbox with 501 gears.

Table 15: Torque loss on test gearbox with 501 gears [N.m].

Speed [rpm]	Load Stage	PAOR	PAOC	PAOM	PAOX	MINR	MINE	PAGD	ESTR
200	k1	0.77	1.11	0.67	0.44	0.59	0.87	0.79	1.15
	k5	1.40	2.27	1.83	1.15	1.49	1.44	1.46	1.05
	k7	2.04	3.72	2.56	2.17	2.75	2.88	1.84	2.10
	k9	2.72	3.55	2.65	3.72	4.46	3.84	2.47	4.48
400	k1	0.99	1.21	0.77	0.48	0.71	0.96	0.89	1.28
	k5	1.45	2.31	1.79	1.21	1.63	1.56	1.42	1.14
	k7	2.04	3.68	2.45	2.25	2.70	2.75	1.81	2.03
	k9	3.42	3.84	2.60	3.62	4.23	3.41	2.45	3.97
1200	k1	2.55	2.79	2.38	0.61	0.98	1.33	1.31	1.57
	k5	2.76	4.03	2.86	1.65	1.96	1.92	1.98	1.55
	k7	3.26	4.26	3.62	2.59	2.88	3.11	2.26	2.29
	k9	4.06	5.30	3.65	3.44	4.17	3.47	2.70	3.86

Table 16: Test gearbox efficiency with 501 gears.

Speed [rpm]	Load stage	PAOR	PAOC	PAOM	PAOX	MINR	MINE	PAGD	ESTR
200	k1	0.830	0.751	0.854	0.908	0.872	0.808	0.826	0.742
	k5	0.987	0.978	0.982	0.989	0.986	0.986	0.986	0.990
	k7	0.990	0.981	0.987	0.989	0.986	0.985	0.991	0.989
	k9	0.992	0.989	0.992	0.988	0.986	0.988	0.992	0.986
400	k1	0.778	0.723	0.831	0.905	0.845	0.784	0.799	0.707
	k5	0.986	0.978	0.983	0.988	0.984	0.985	0.986	0.989
	k7	0.990	0.981	0.988	0.989	0.986	0.986	0.991	0.990
	k9	0.989	0.988	0.992	0.989	0.987	0.989	0.992	0.988
1200	k1	0.357	0.284	0.390	0.886	0.777	0.683	0.688	0.617
	k5	0.973	0.961	0.972	0.984	0.981	0.982	0.981	0.985
	k7	0.983	0.978	0.982	0.987	0.985	0.984	0.989	0.988
	k9	0.987	0.983	0.989	0.989	0.987	0.989	0.992	0.988

4.3.2.1 *PAO´s performance*

For load stage k1 (Figure 31), representative of the no-load losses, the PAOX present the lowest value of torque loss, while the highest torque loss is present by PAOC oil.

The same behaviour is observed at load stage k5 (Figure 32).

For load stage k7 (Figure 33), the PAOC oil presents the worst performance, while the oil that presents the best performance changes with the input speed, being PAOR in the range speed 200 to 400 rpm and PAOX at a speed of 1200 rpm.

In load stage k9 (Figure 34) at 200 rpm, PAOM and PAOR had the lowest torque loss, while PAOX and PAOC had the higher torque loss. At 400 rpm the PAOM and PAOC oils are stil distinguished with the lower and the higher torque loss, respectively. For an input speed of 1200 rpm the PAOX oil presents the best performance, slightly better than PAOM.

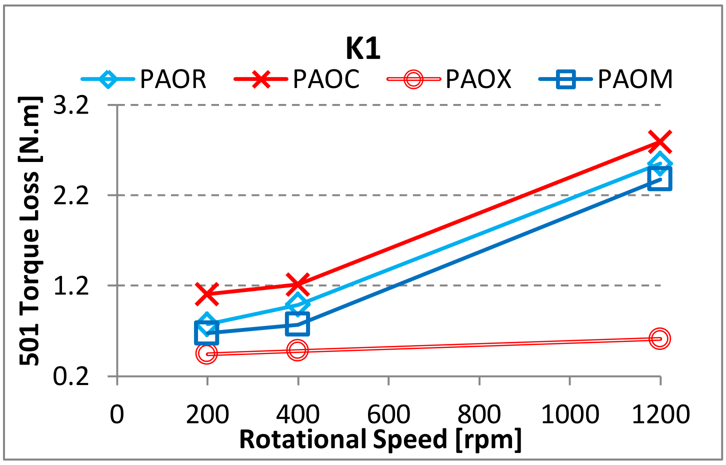

Figure 31: Torque loss of 501 gear at load stage K1 lubricated with PAO's.

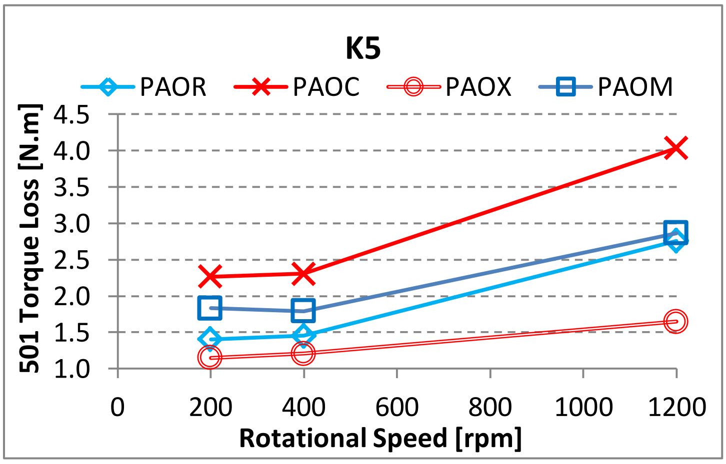

Figure 32: Torque loss of 501 gear at load stage K5 lubricated with PAO's.

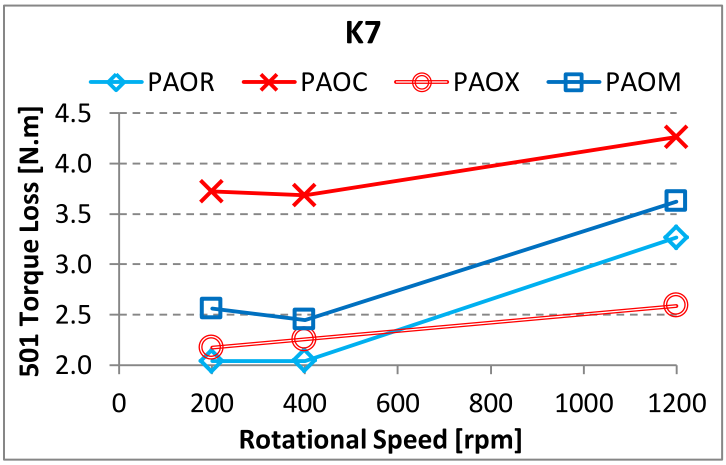

Figure 33: Torque loss of 501 gear at load stage K7 lubricated with PAO's.

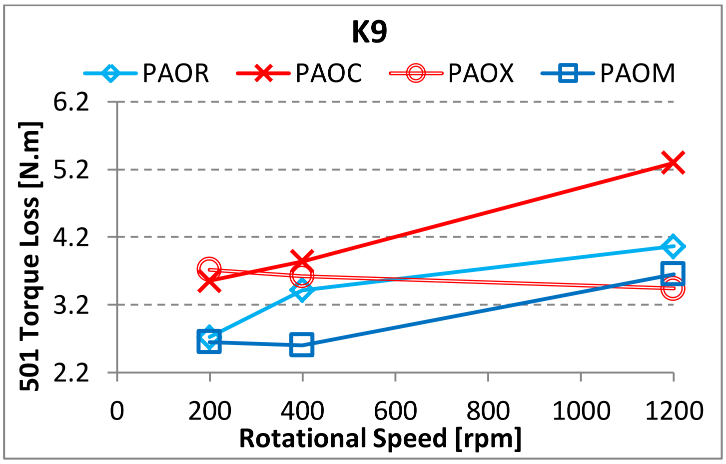

Figure 34: Torque loss of 501 gear at load stage K9 lubricated with PAO's.

With 501 gear design and, in the no-load losses representative stage (Figure 35), PAOX present the high efficiency and PAOC present the lowest efficiency.

Increasing the load, load stage k5 (Figure 36), PAOC distinguishes itself with lower efficiency than others for all speed range. The best efficiency is shown by PAOX for all speed range.

For k7 load stage (Figure 37 at a speed range from 200 to 400 the best efficiency is present by PAOR while at higher speed PAOX has higher efficiency.

For the highest load (Figure 38) at low speeds the difference between PAOM and PAOR and between PAOX and PAOC is very small presenting the high and the low efficiency, respectively. At 400 rpm PAOM and PAOC distinguished than others with highest and lowest efficiency, respectively. At 1200 rpm PAOC and PAOX present best and worst efficiency, respectively than others.

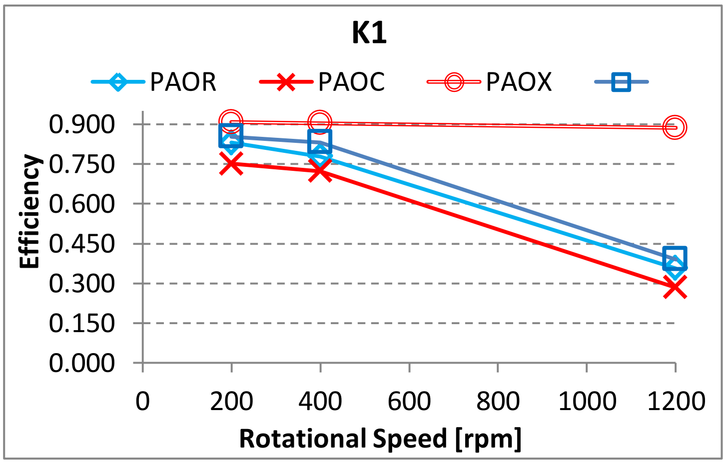

Figure 35: Efficiency of 501 gear at load stage K1 lubricated with PAO's.

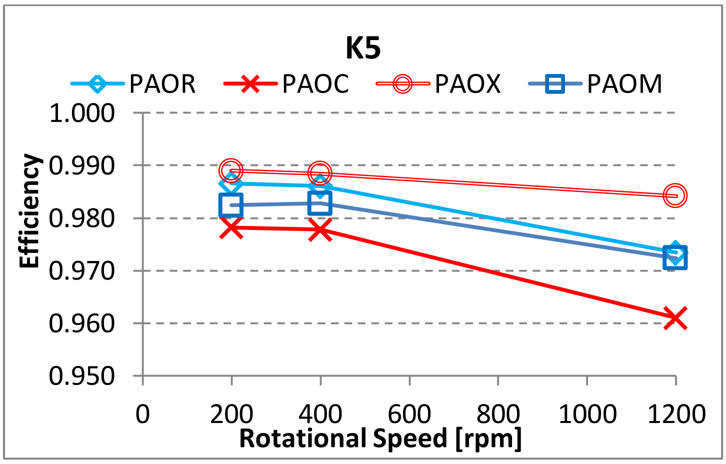

Figure 36: Efficiency of 501 gear at load stage K5 lubricated with PAO's.

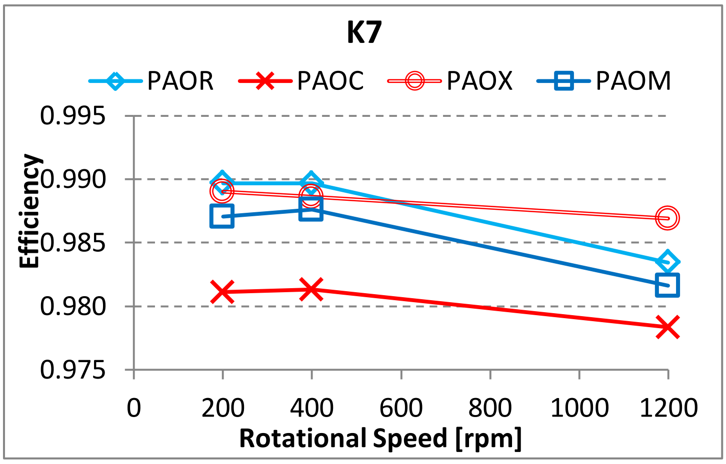

Figure 37: Efficiency of 501 gear at load stage K7 lubricated with PAO's.

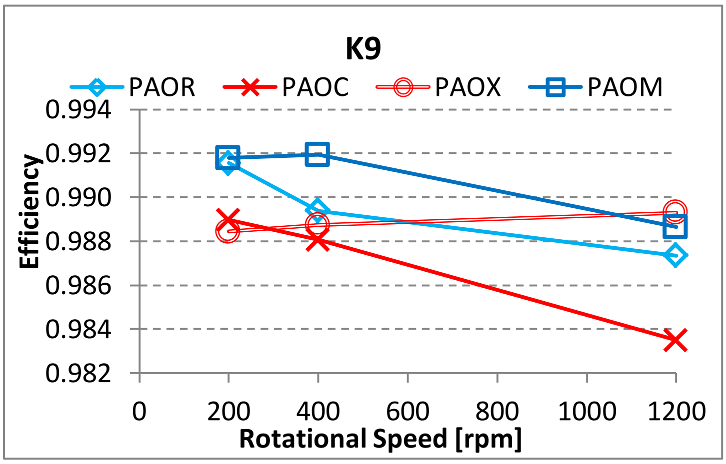

Figure 38: Efficiency of 501 at gear load stage k9 lubricated with PAO's.

4.3.2.2 *PAOR, MINR, MINE, PAGD and ESTR performance*

For load stage k1 (Figure 39) MINR has the best performance presenting lower torque loss than others for all speed range. At the speed range from 200 to 400 rpm the MINE, PAGD and PAOR present approximately the same value, while ESTR presents the highest torque loss. At 1200 rpm, with a large difference to the others oils, PAOR presents the highest torque loss.

In k5 load stage (Figure 40) ESTR has the best performance presenting lower torque loss than all the others for all the speed range. MINR, MINE, PAGD and PAOR present approximately the same value. At 1200 rpm PAOR present the high torque loss.

At load stage k7 (Figure 41) and for all speed range PAGD present the lowest torque loss. At 1200 rpm, the ESTR oil presents approximately the same value as the PAGD. The mineral oils present the worst at the speeds of 200 to 400 rpm, while PAOR was the highest torque loss at 1200 rpm.

At load stage k9 (Figure 42) the PAGD oil has the best performance. The MINR oil has the worst performance, presenting the highest torque loss. The ESTR has worsened its behaviour with the increase of load while the MINE has improved its performance with the increase of load.

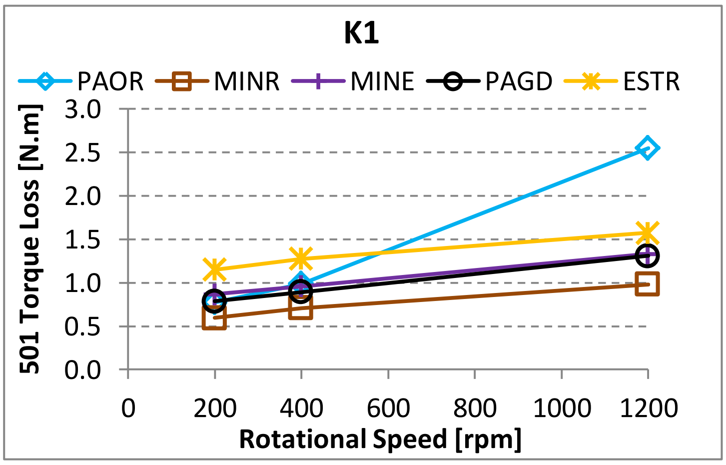

Figure 39: Torque loss of 501 gear at load stage K1.

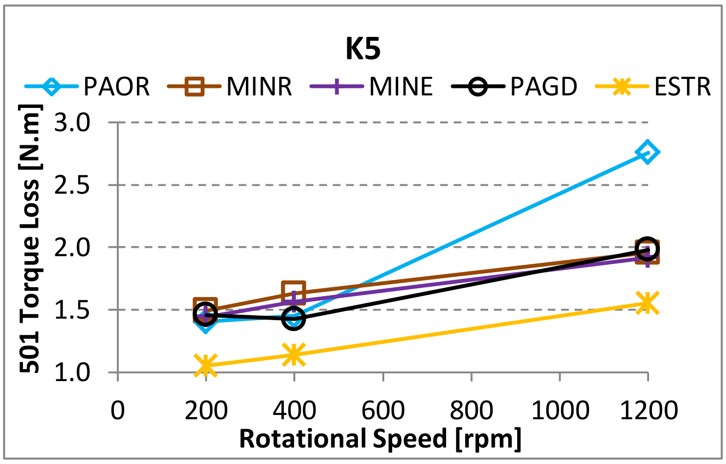

Figure 40: Torque loss of 501 gear at load stage K5.

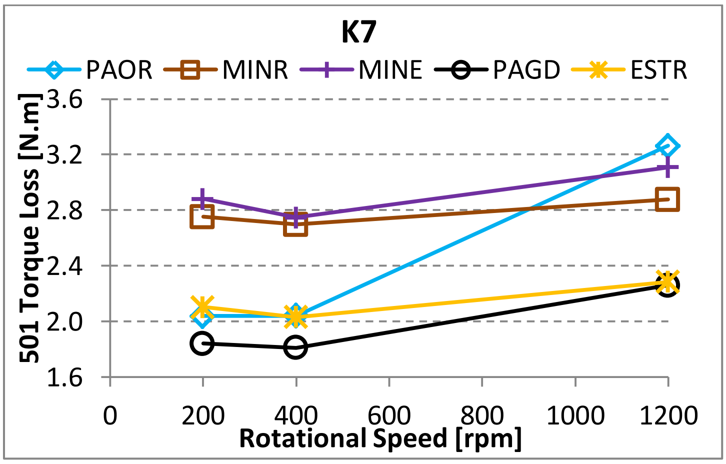

Figure 41: Torque loss of 501 gear at load stage K7.

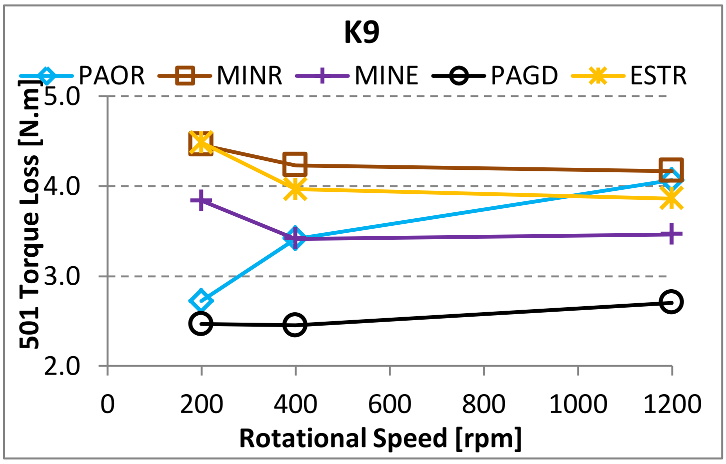

Figure 42: Torque loss of 501 gear at load stage K9.

For k1 load stage (Figure 43) MINR present the highest efficiency, independently of the speed. At the speeds of 200 and 400 rpm, the ESTR oil presents low efficiency while PAOR, MINE and PAGD had approximately the same efficiency. At high speeds PAOR presents the lowest efficiency, PAGD and MINE had approximately the same efficiency value.

Increasing the load, to stage k5 (Figure 44), changes the relative behaviour of the oils, with the ESTR presenting the best efficiency and the MINR presenting the lower efficiency, except for 1200 rpm where the PAOR has the lowest efficiency. The efficiency differences between PAGD, MINR and MINE are very small at this load stage.

At load stage K7 (Figure 45), PAGD presents the highest efficiency for all the speed range, although ESTR has similar values.

In load stage k9 (Figure 46), the PAGD oil presents the highest efficiency for all speeds. at this load stage the MINR presents the lowest efficiency, closely followed by ESTR.

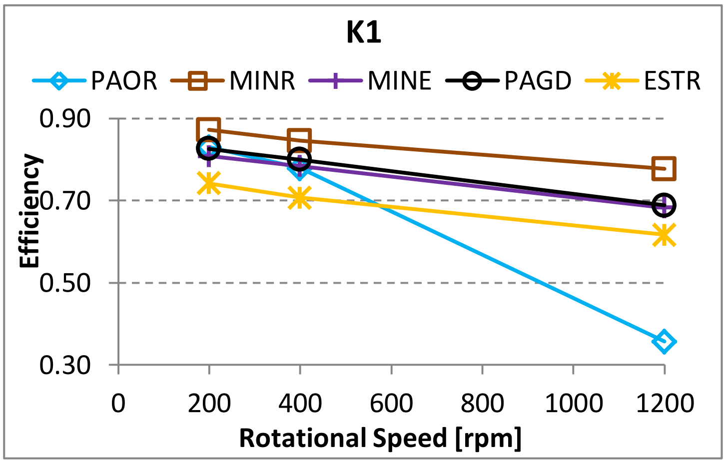

Figure 43: Efficiency of 501 gear at load stage K1.

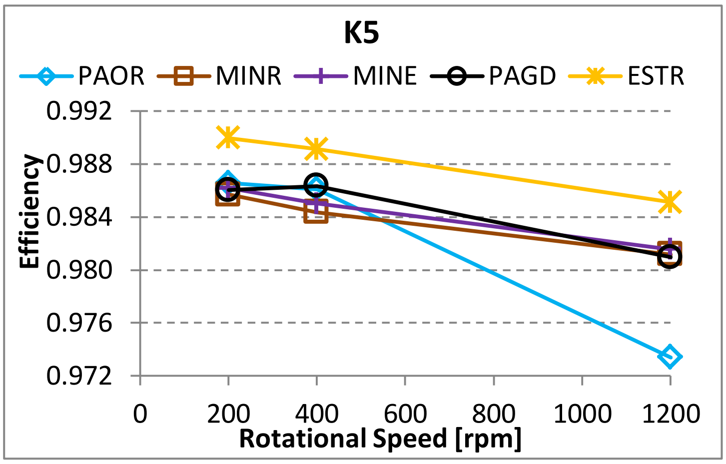

Figure 44: Efficiency of 501 gear at load stage K5.

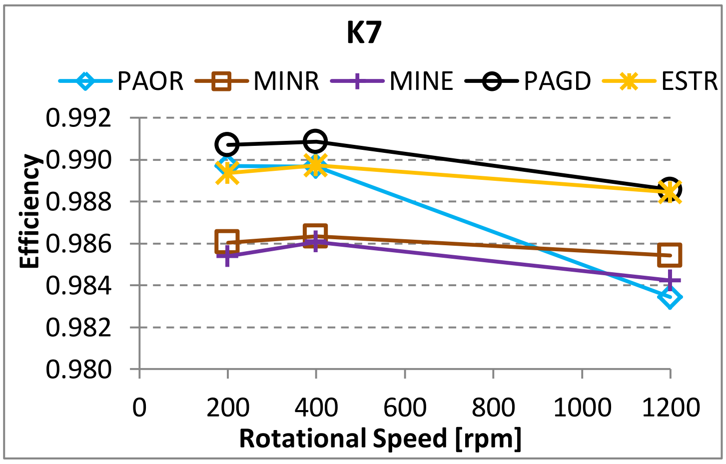

Figure 45: Efficiency of 501 gear at load stage K7.

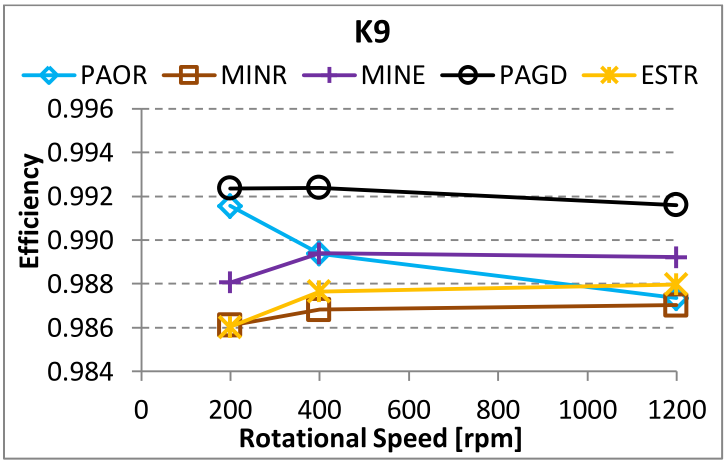

Figure 46: Efficiency of 501 gear at load stage K9.

4.3.2.3 *Results discussion and comparasion*

For load stage k1 the PAO oils have approximately the same viscosity, so viscosity only by itself should not be the explanation for the differences observed, even though this load stage is performed at very low load to simulate no-load losses.

The oil that presented the highest losses was always the PAOC, whatever the speed and load.

The PAOM lubricant has a quite bad behaviour for low loads but for high loads its torque losses became the best of the PAO group. The PAOX has the opposite behaviour, displaying the lowest losses for the lower loads and the highest losses for the K9 load stage.

The PAOR has consistently one of the best behaviours on the group.

The oil MINR has the lower torque loss for the no-load stage (K1) but for higher loads it has in general the highest load losses.

The PAGD oil has a high torque loss for the no-load stage, as expected due to its high viscosity and high density, but for the higher load stages it become the oil with the lowest torque loss.

The MINE oil displays, comparatively, quite high torque losses for the lower load stages but it improves its relative behabiour with the increase of load.

The ESTR has a good behaviour for the intermediate loads, but for the highest load and for the no load conditions it displays high torque losses.

Table 17 displays the bulk and flash temperature calculated for the PAO base oils at all load stages.

Table 17: Flash and bulk temperature of PAO's base oils for 501 gear and for all tests performed.

	PAOR		PAOX		PAOC		PAOM	
Load stage	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}
K1	93.19	0.30	95.13	0.30	93.22	0.30	95.46	0.30
K5	104.78	11.68	105.68	11.71	104.79	11.70	105.67	11.66
K7	117.62	25.11	117.68	25.19	115.93	25.16	117.51	25.06
K9	133.86	45.04	134.94	45.18	134.18	45.12	134.05	44.95

The oils specific film thickness (Figure 47) does not have large variations because their viscosity and piezoviscosity is approximately the same.

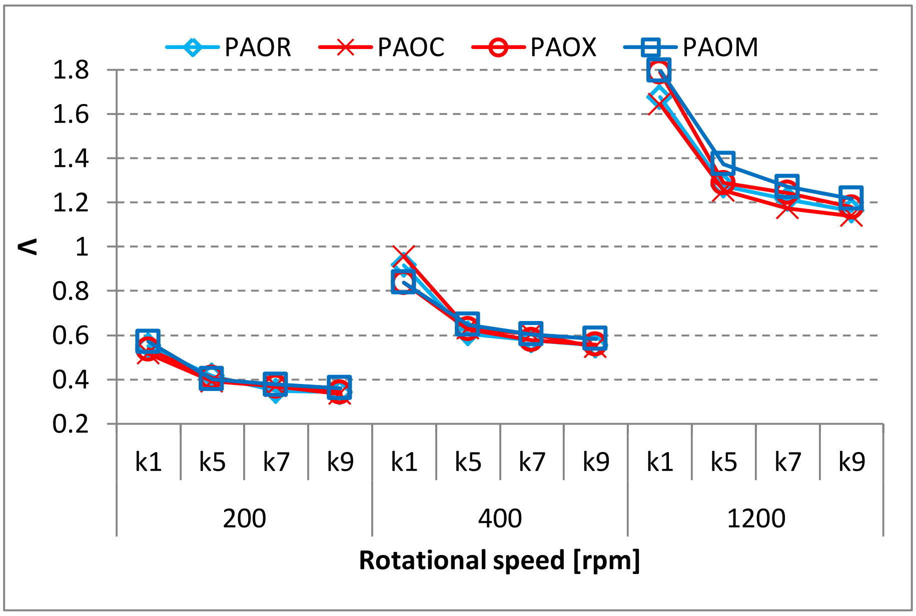

Figure 47: Specific film thickness for 501 gears lubricated with PAO's.

Table 18 displays the bulk and flash temperature calculated for the PAO oils at different load stages.

Table 18: Flash and bulk temperature on 501 gears for each test performed.

	PAOR		MINE		MINR		PAGD		ESTR	
Load stage	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}	T_{bulk}	T_{flash}
K1	93.19	0.30	96.27	0.30	96.27	0.30	94.51	0.30	96.43	0.30
K5	104.78	11.68	105.92	11.65	106.20	11.67	104.45	11.73	105.81	11.70
K7	117.62	25.11	117.20	25.05	117.22	25.10	117.51	25.23	117.18	25.17
K9	133.86	45.04	133.70	44.93	133.91	45.01	134.26	45.25	134.20	45.14

The specific film thickness of MINR, ESTR and PAOR are approximately the same. PAGD and MINE have the larger specific film thickness of the group, besides PAGD having higher viscosity and smaller piezoviscosity than MINE.

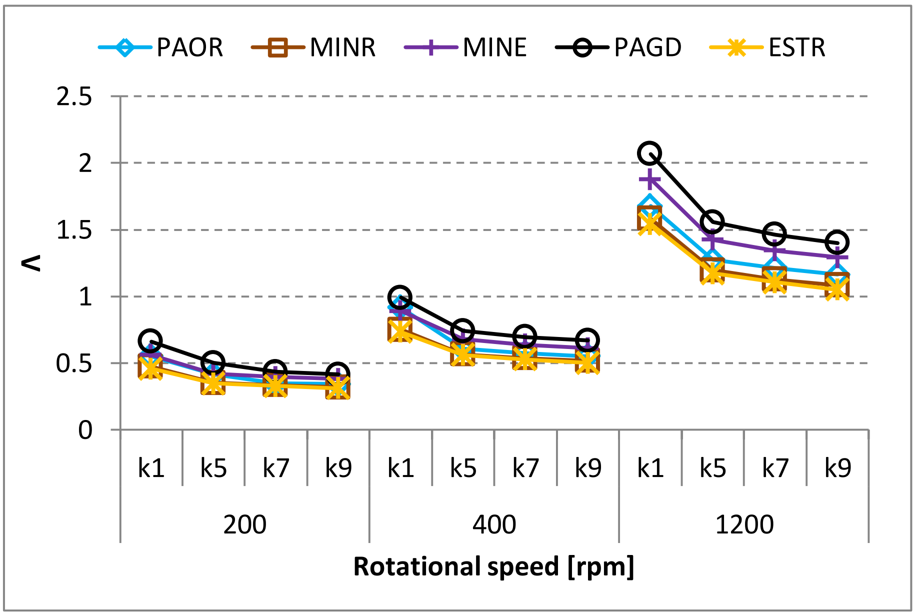

Figure 48: Specific film thickness for 501 gears.

4.3.2.3.1 Comparison of the gear geometries C40 and 501

In this section the behaviour of the two gear designs tested will be compared. This comparison is not straight forward, once the contact pressures are quite different and the surface roughness is also quite different, nevertheless the operating conditions are the same (input speed, input torque and oil temperature).

4.3.2.3.1.1 PAO's

The comparison of the gear geometries shows three different behaviours dependent on the lubricants.

PAOR (Figure 49) and PAOM (Figure 52) have similar behaviour, with both geometries having similar torque losses for the speeds of 200 and 400 rpm, although the 501 geometry has lower power losses for the highest load (load stage K9). The C40 spur gears at 1200 rpm had lower torque loss, a reflex of the higher no load losses on the spur geometry.

The oil PAOC (Figure 50) always displayed higher torque loss with the helical gear 501.

The oil PAOX (Figure 51) always displayed lower torque loss with 501 gear.

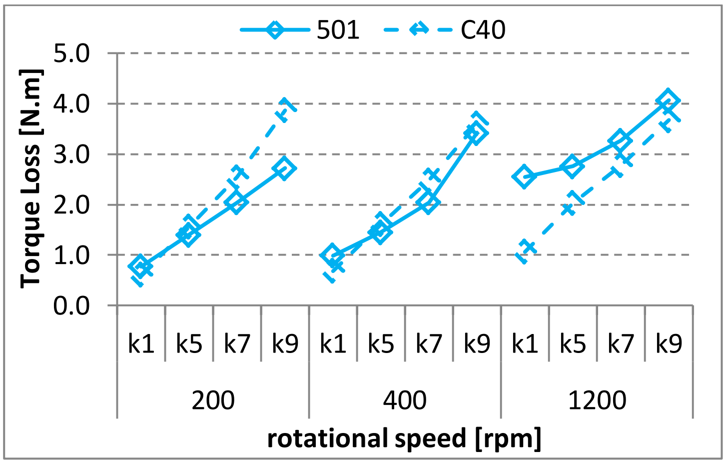

Figure 49: Torque loss comparison of C40 and 501 gear geometries using PAOR oil.

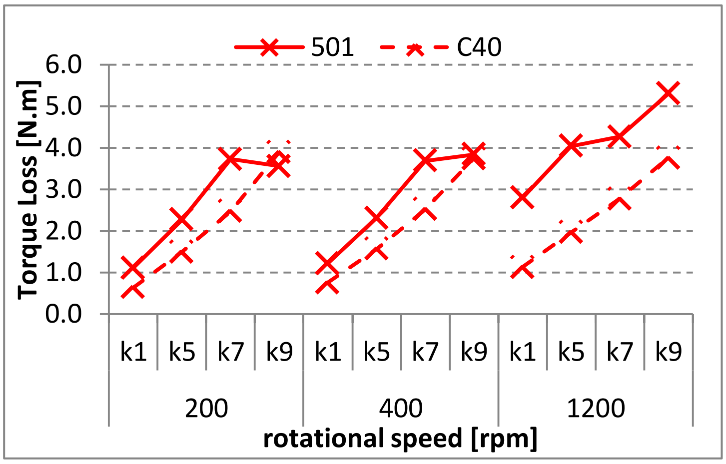

Figure 50: Torque loss comparison of C40 and 501 gear geometries using PAOC oil.

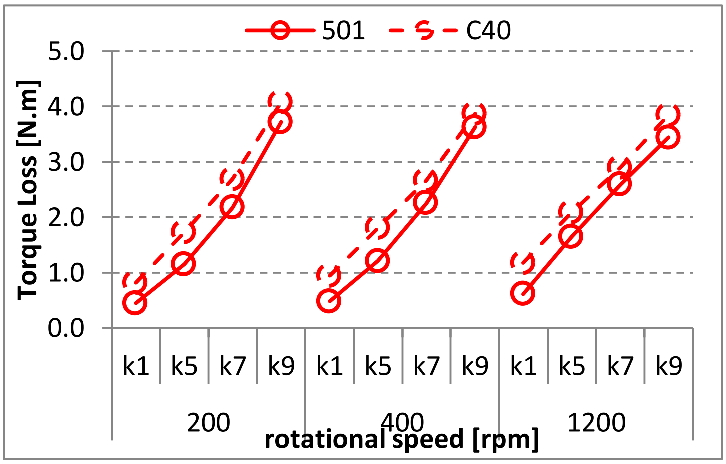

Figure 51: Torque loss comparison of C40 and 501 gear geometries using PAOX oil.

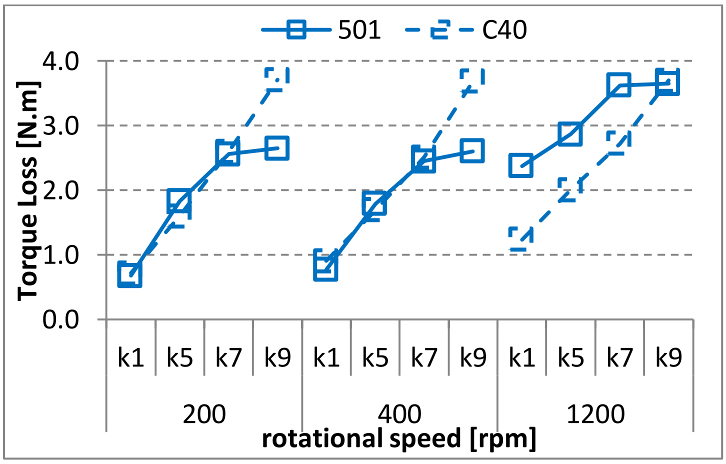

Figure 52: Torque loss comparison of C40 and 501 gear geometries using PAOM oil.

The comparison of the gear geometries efficiency for each oil tested is represented in Figure 53 till Figure 56. These figures show that the biggest efficiency differences are observed for low load and high speed, where improvements up to 2% are observed. For high loads, the maximum improvement due to geometry is smaller than 0.5%.

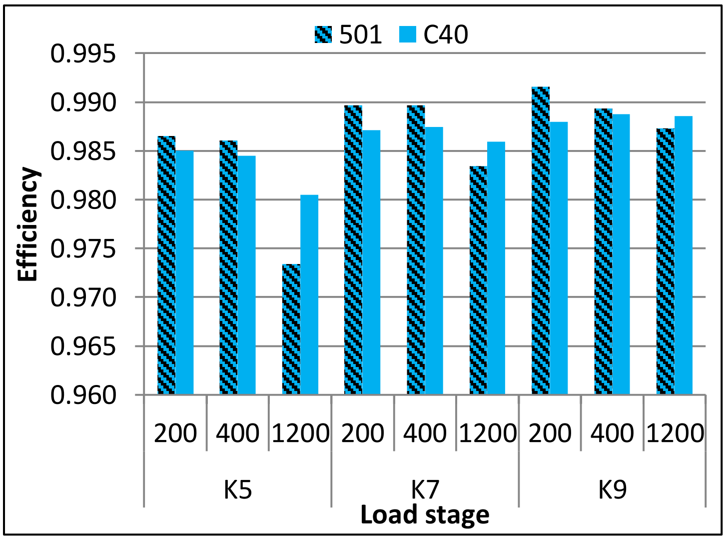

Figure 53: Efficiency comparison of C40 and 501 gear geometries using PAOR oil.

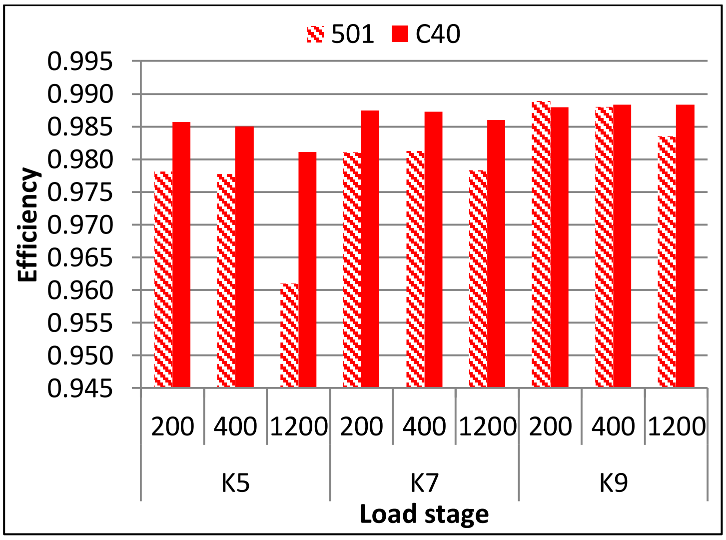

Figure 54: Efficiency comparison of C40 and 501 gear geometries using PAOC oil.

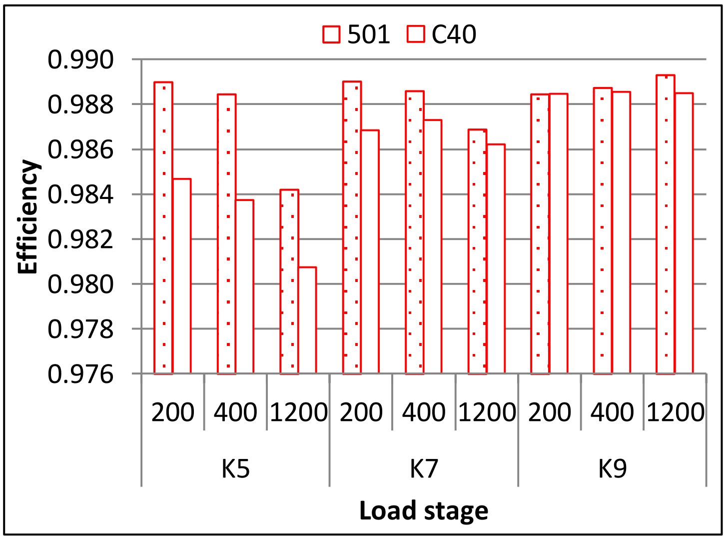

Figure 55: Efficiency comparison of C40 and 501 gear geometries using PAOX oil.

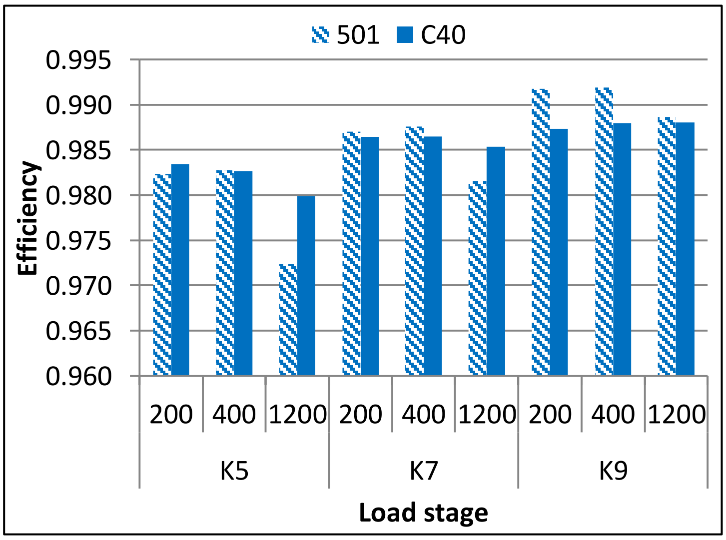

Figure 56: Efficiency comparison of C40 and 501 gear geometries using PAOM oil.

4.3.2.3.2 MINR, MINE, PAGD and ESTR

The MINR oil presented a torque loss similar for both 501 and C40 gear geometry (Figure 57) With the oil MINE within the speed range 200 to 400 rpm, both geometries displayed aproximatelly the same torque loss, although fo the input speed of 1200 rpm the geometry 501 has a much lower torque loss, as displayed in Figure 58. Figure 59 represents the torque loss for PAGD oil, and it can be observed that the 501 gear geometry has lower torque loss than the spur geometry, except at load stage K1 where both geometries have the same losses. The oil ESTR has a curious behaviour, with the gear 501 displaying lower power losses for the intermediate loads and worst behaviour for the lowest and highest load (Figure 60).

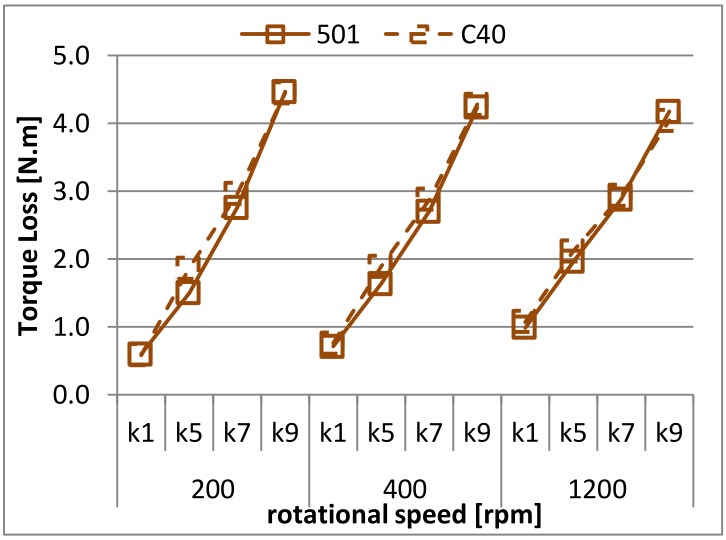

Figure 57: Torque loss comparison of C40 and 501 gear geometries using MINR oil.

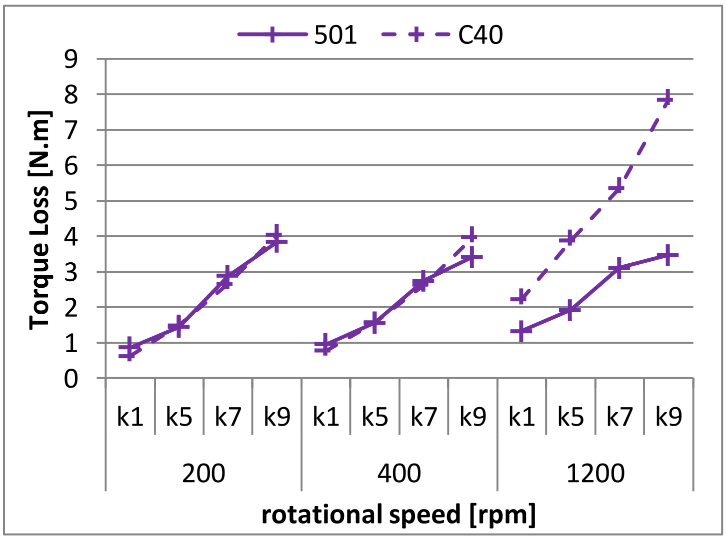

Figure 58: Torque loss comparison of C40 and 501 gear geometries using MINE oil.

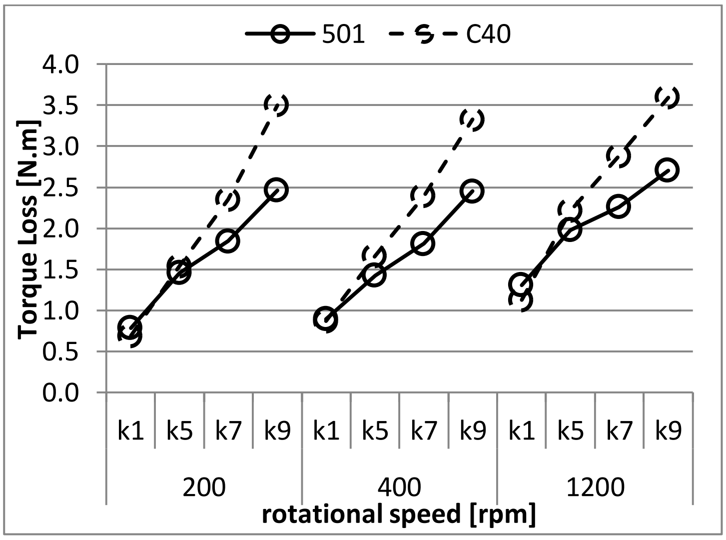

Figure 59: Torque loss comparison of C40 and 501 gear geometries using PAGD oil.

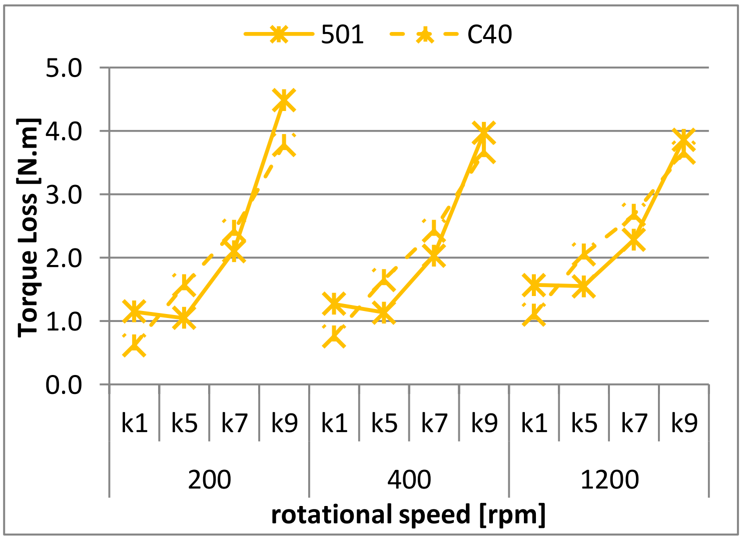

Figure 60: Torque loss comparison of C40 and 501 gear geometries using ESTR oil.

Figure 61 until Figure 64 display the efficiency comparison of the gear geometries 501 and C40 for the lubricants MINR, MINE, PAGD and ESTR, respectively. These figures show that the 501 gear geometry has in general better efficiency than the spur geometry C40, except for the ESTR oil at the highest load stage. The improvements obtained can reach 0.5%.

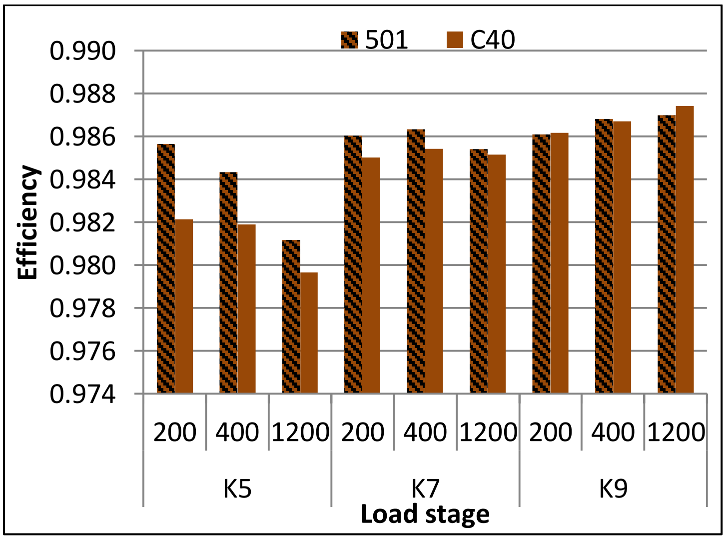

Figure 61: Efficiency comparison of C40 and 501 gear geometries using MINR oil.

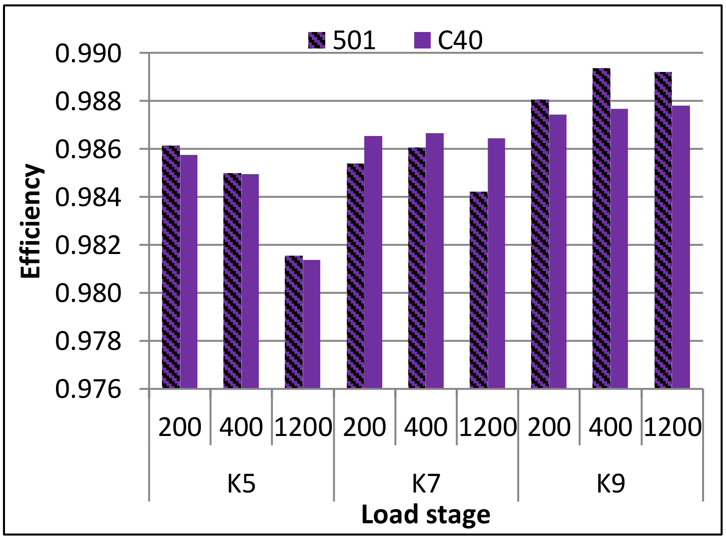

Figure 62: Efficiency comparison of C40 and 501 gear geometries using MINE oil.

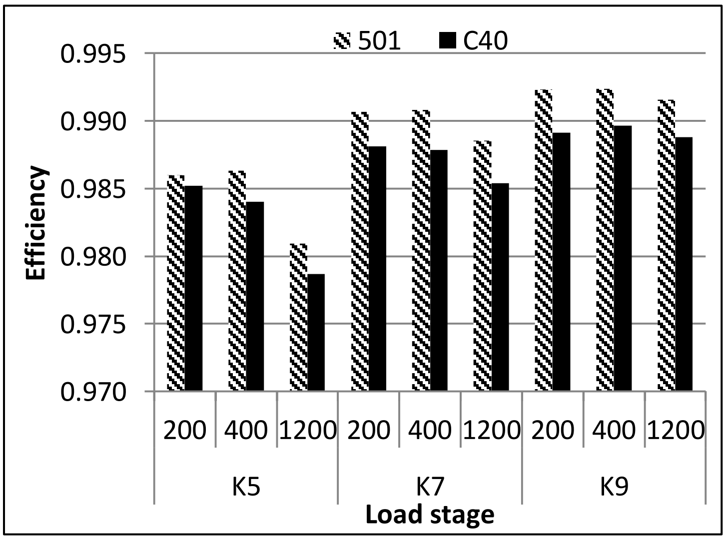

Figure 63: Efficiency comparison of C40 and 501 gear geometries using PAGD oil.

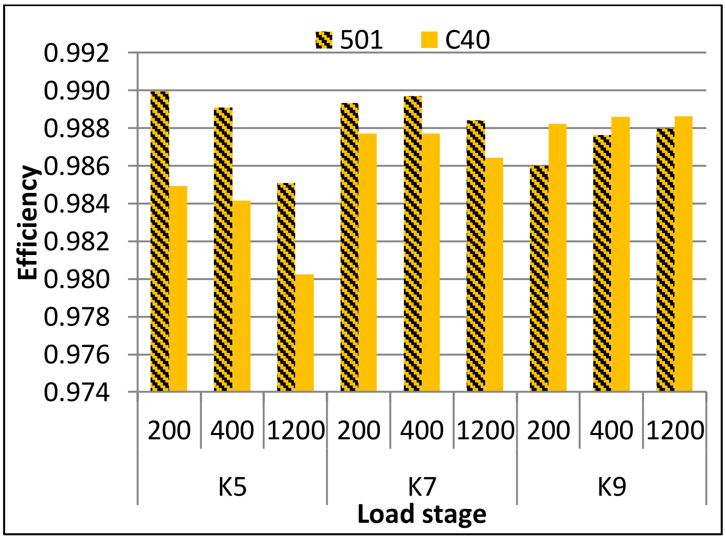

Figure 64: Efficiency comparison of C40 and 501 gear geometries using ESTR oil.

4.3.3 951 gear design

Such as for C40 and 501 gear design, the discussions of the oils performances will be made. For this gear design has only been tested PAOR, MINR and PAGD oils because there was no more time to test other oils. The torque loss is present in Table 19 and the test gearbox efficiency in Table 20.

Table 19: Torque loss on test gearbox with 951 gears [N.m].

Speed [rpm]	Load Stage	PAOR	MINR	PAGD
200	k1	0.95	1.31	1.08
	k5	0.75	0.89	0.75
	k7	1.09	1.58	1.10
	k9	1.87	3.52	1.23
400	k1	1.17	1.35	1.42
	k5	0.98	1.13	1.11
	k7	1.31	1.74	1.31
	k9	2.04	3.47	1.51
1200	k1	2.75	1.82	3.73
	k5	2.58	1.61	3.23
	k7	2.78	2.14	3.48
	k9	2.41	3.71	3.63

Table 20: : Test gearbox efficiency with 951 gears.

Speed [rpm]	Load stage	PAOR	MINR	PAGD
200	k1	0.789	0.707	0.755
	k5	0.993	0.992	0.993
	k7	0.995	0.992	0.994
	k9	0.994	0.989	0.996
400	k1	0.736	0.690	0.667
	k5	0.991	0.989	0.989
	k7	0.993	0.991	0.993
	k9	0.994	0.989	0.995
1200	k1	0.305	0.553	0.027
	k5	0.975	0.985	0.969
	k7	0.986	0.989	0.982
	k9	0.993	0.988	0.989

4.3.3.1 ***PAOR, PAGD and MINR performance***

For load stage k1 (Figure 65), no-load losses representative, PAOR presents best performed at speed range from 200 to 400 rpm and at 1200 rpm MINR has the best performance. At 200 rpm MINR present the worst performance. At 400 rpm MINR and PAGD has approximately the same torque loss. At 1200 rpm, with a large difference to the other oils, PAGD presents the highest torque loss..

Increasing the load (k5 load stage) (Figure 66), the dispositions is approximately the same that in k1 load stage. At speed range from 200 to 400 rpm the MINR, PAGD and PAOR have approximately the same toque loss. At higher speed (1200 rpm) MINR present the lowest torque loss and PAGD present the highest torque loss.

At load stage k7 (Figure 67) and at speed range from 200 to 400 rpm PAGD and PAOR present approximately the same value, while MINR presents the highest torque loss. At 1200 rpm the PAGD and MINR disguised itself with highest and lowest torque loss, respectively.

At higher load, k9 load stage (Figure 68), and for range speed 200 to 400 rpm, PAGD present the lowest torque loss. At higher speed (1200 rpm) PAOR distinguished with low torque loss. The highest torque loss is presented by MINR for all range speed.

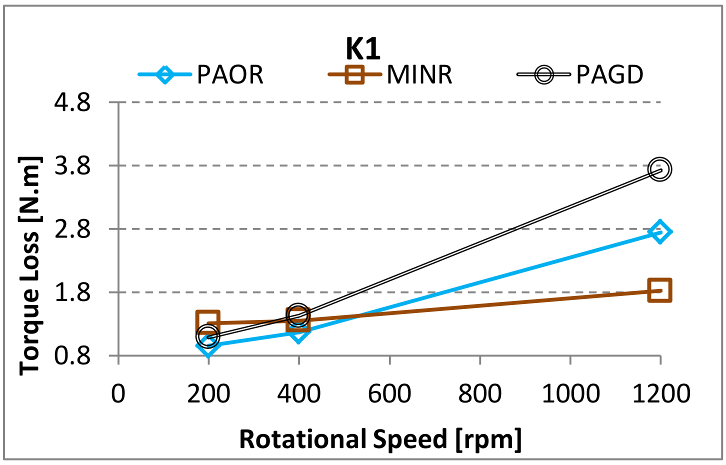

Figure 65: Torque loss of 951 gear at load stage K1.

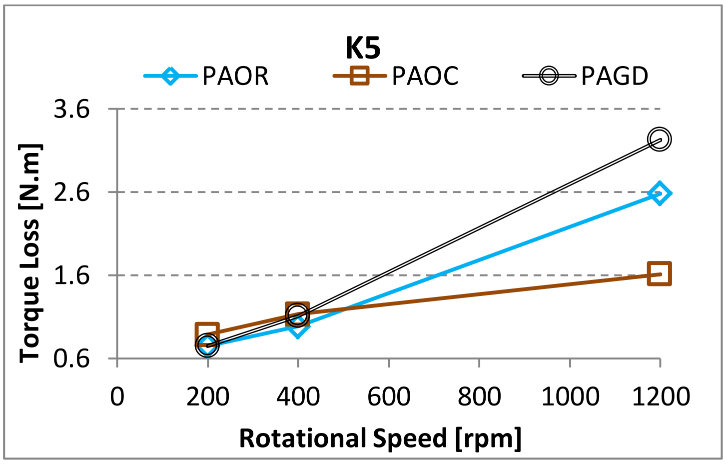

Figure 66: Torque loss of 951 gear at load stage K5.

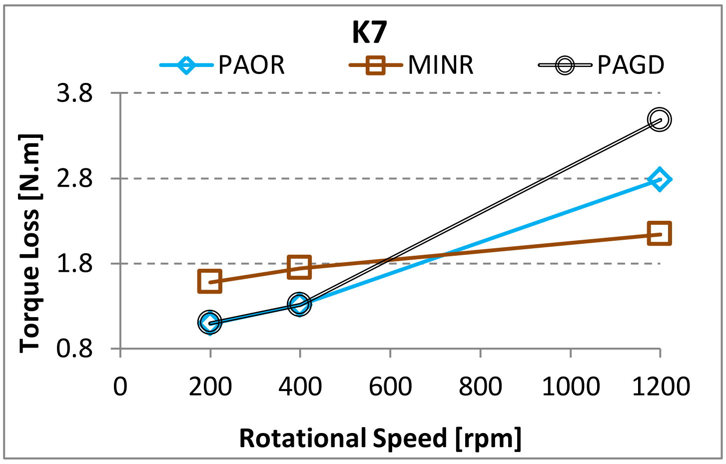

Figure 67: Torque loss of 951 gear at load stage K7.

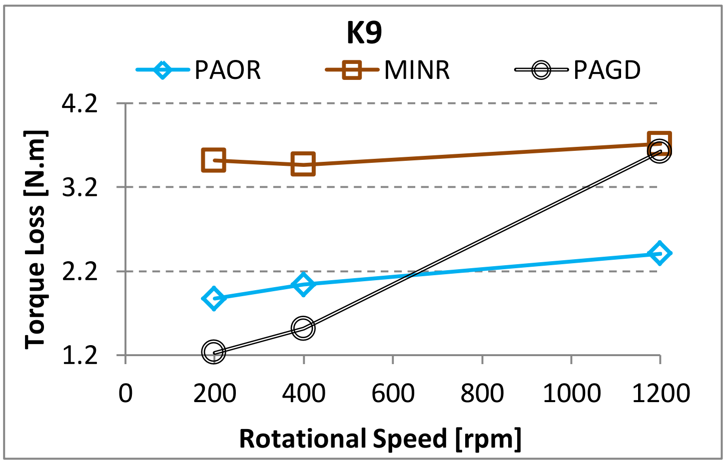

Figure 68: Torque loss of 951 gear at load stage K9.

At k1 load stage (Figure 69) and at speed range from 200 to 400 rpm PAOR has the highest efficiency. MINR at 200 rpm and PAGD at range speed 400 to 1200 present the worst efficiency. At higher speed MINR present the best efficiency.

For k5 load stage (Figure 70) at 200 rpm PAGD and PAOR present the highest efficiency with the same value and MINR the lowest efficiency. Increasing the speed (400 rpm), the POAR present a good efficiency then others and MINR and PAGD present bed efficiency with the same value. At higher speed PAGD and MINR present again the highest and lowest efficiency, respectively.

For a considerable load, k7 (Figure 71) load stage, and at range speed 200 to 400 rpm MINR isolate itself with bed efficiency and PAGD and PAOR the highest efficiency with the same value. At 1200 rpm MINR present the best efficiency and PAGD the worst efficiency.

For high loads (k9 load stage) MINR continuous with bed efficiency and PAGD, except for high speeds (1200 rpm), present the better efficiency (Figure 72). At 1200 rpm PAOR back to have a good performance with higher efficiency then others.

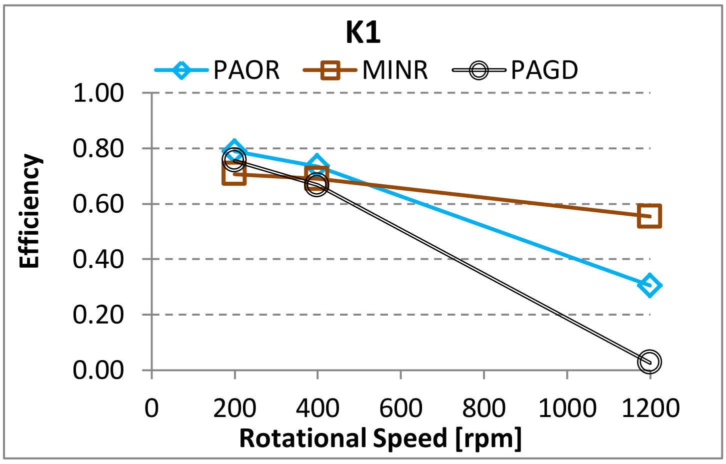

Figure 69: Efficiency of 951 gear load stage K1.

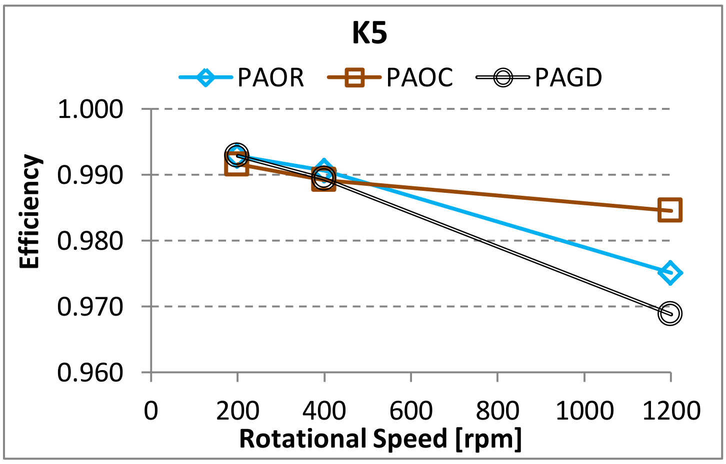

Figure 70: Efficiency of 951 gear load stage K5.

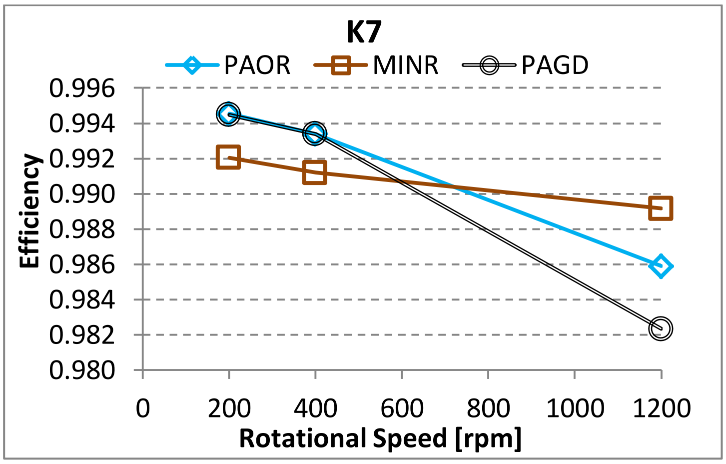

Figure 71: Efficiency of 951 gear load stage K7.

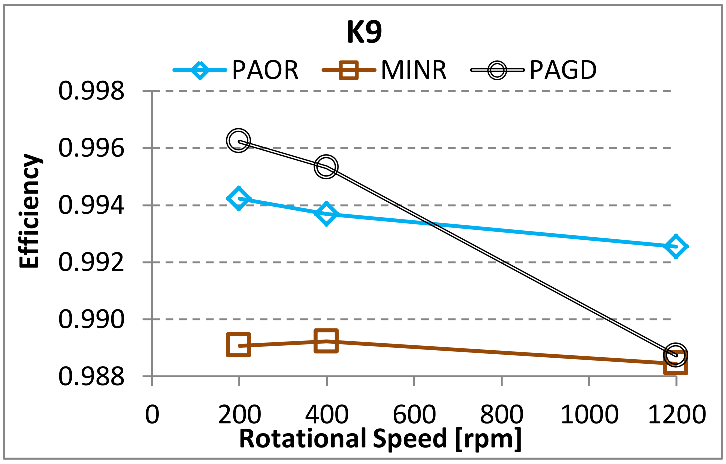

Figure 72: Efficiency of 951 gear load stage K9.

4.3.3.2 *Results discussion and comparison*

For load stage k1 (Figure 65), and despite being representative of the no-load losses, the good performance of PAOR and the MINR bed performance at 200 rpm can be due of the good conditions of the lubricant film creation. At range speed 400 to 1200 rpm PAGD presents the highest torque loss and thus the viscosity dependency is notorious. At high speeds the oils torque loss is disposed second the viscosity.

In load stage k5 (Figure 66), is approximately similar with k1 load stage. The highest toque present by MINR at 200 to 400 rpm and the PAOR good performance for all speed range can be due the regime lubrication and at 1200 the viscosity dependency is again notorious.

The torque loss in load stage k7 (Figure 67), PAGD and PAOR have good performance at range speed 200 to 400 rpm. This improves can be due the good film formation conditions. The highest torque presents by MINR for all range speed eliminates the viscosity dependency and can be due of the bed conditions of film formation. Such in k1 and k5 load stage at 1200 rpm, in k7 load stage is visible a viscosity dependency.

The good performance of PAGD and the bed performance of MINR in k9 load stage (Figure 68) eliminate the viscosity dependency.

Table 21 displays the bulk and flash temperature calculated for the PAO oils at different load stages.

Table 21: Flash and bulk temperature on 951 gears for each test performed.

	PAOR		MINR		PAGD	
	Tbulk	Tflash	Tbulk	Tflash	Tbulk	Tflash
K1	96.40	0.30	95.41	0.30	96.78	0.30
K5	106.28	11.68	105.57	11.67	105.90	11.73
K7	117.66	25.11	117.17	25.10	117.43	25.23
K9	134.23	45.04	134.22	45.01	134.48	45.25

Such in 501 gear design, the oils specific film thickness with 951 gear design (Figure 73) are arranged inversely to the piezoviscosity. The highest specific film thickness of PAGD proves the good performance at high loads.

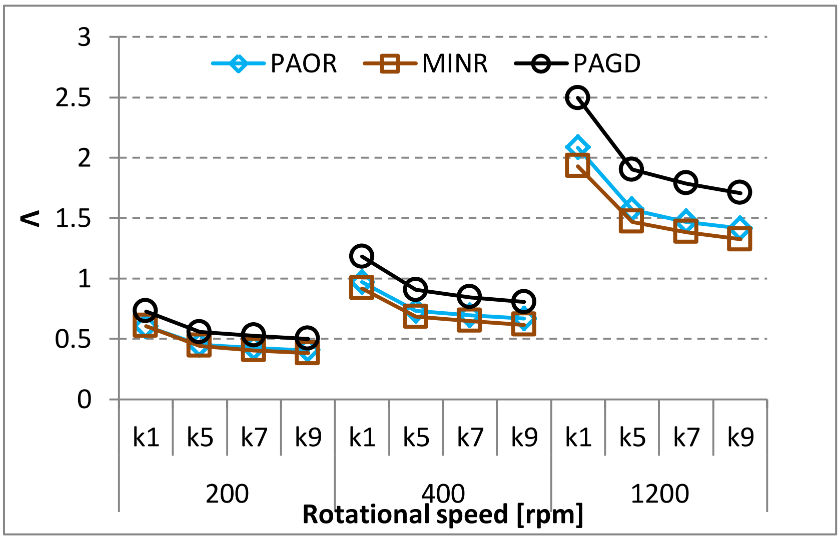

Figure 73: Specific film thickness for 951 gears.

4.3.3.2.1 Comparison of the gear geometries 501 and 951

In this section the behaviour of the 501 gears and 951 gears will be compared. This comparison is not straight forward, once the contact pressures are quite different and the surface roughness is also quite different, nevertheless the operating conditions are the same (input speed, input torque and oil temperature).

The comparison of the gears 951 and 501 torque loss for each oil tested is represented in Figure 74 till Figure 76. The 951 gears design displayed higher torque loss at k1 load stage gears and at others test load stage lower when compared with 501 gears, independently the oil test. So it can conclude that 951 gears are load losses reducers.

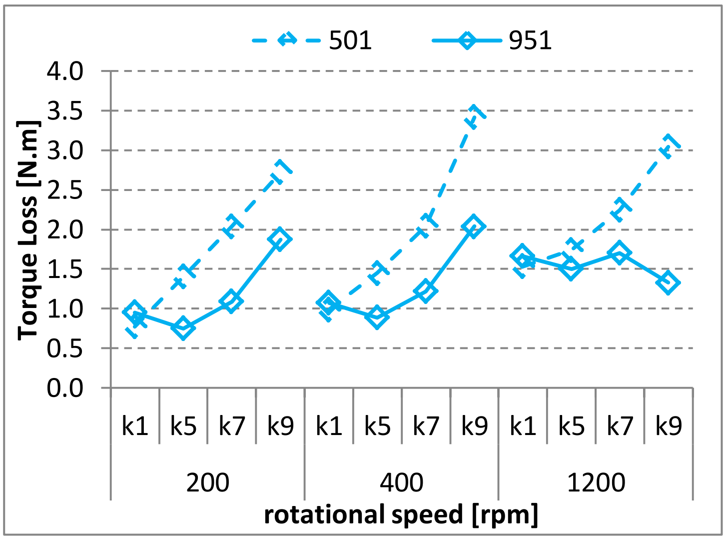

Figure 74: Torque loss comparison of 501 and 951 gear geometries using PAOR oil.

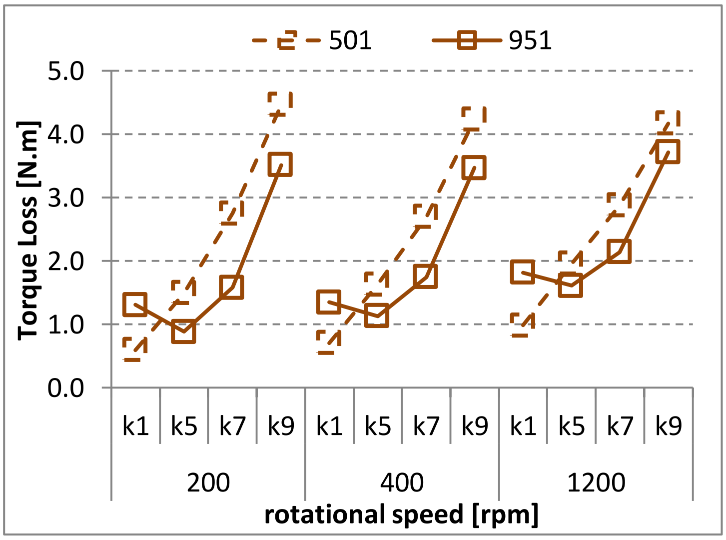

Figure 75: Torque loss comparison of 501 and 951 gear geometries using MINR oil.

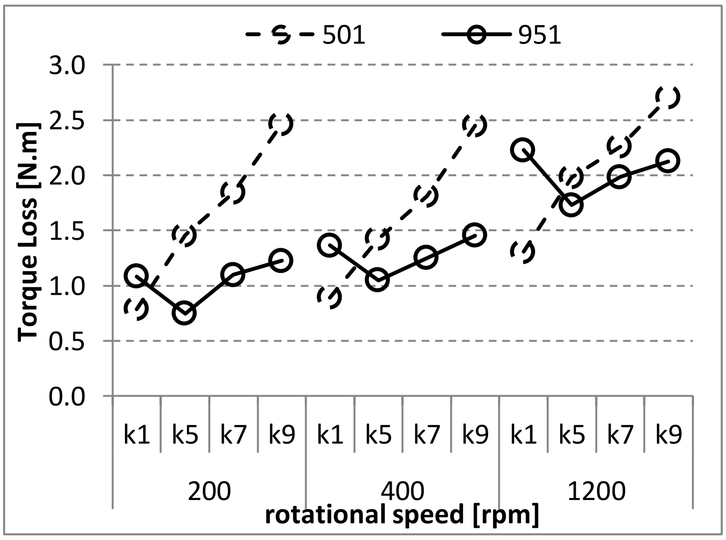

Figure 76: Torque loss comparison of 501 and 951 gear geometries using PAGD oil.

Figure 77 until Figure 79 display the efficiency comparison of the gear geometries 951 and 501 for the test lubricants. 951 gears present better efficiency than 501 gears, except for the PAGD oil at highest speed. The improvements obtained can reach 1.2%.

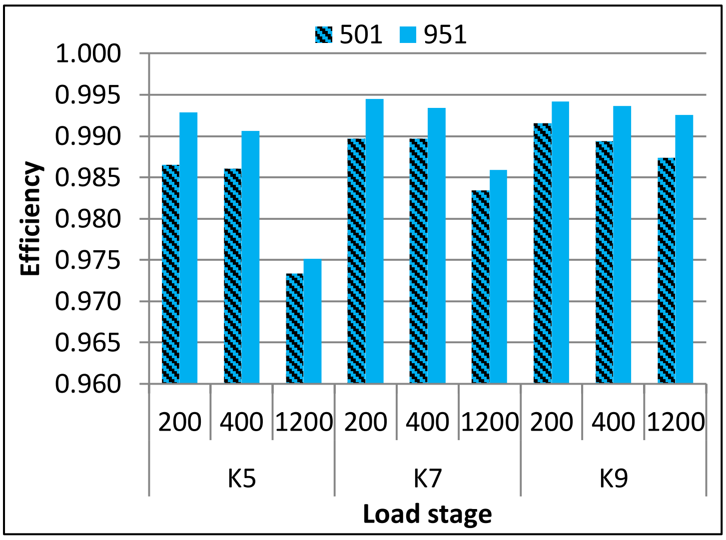

Figure 77: Efficiency comparison of 501 and 951 gear geometries using PAOR oil.

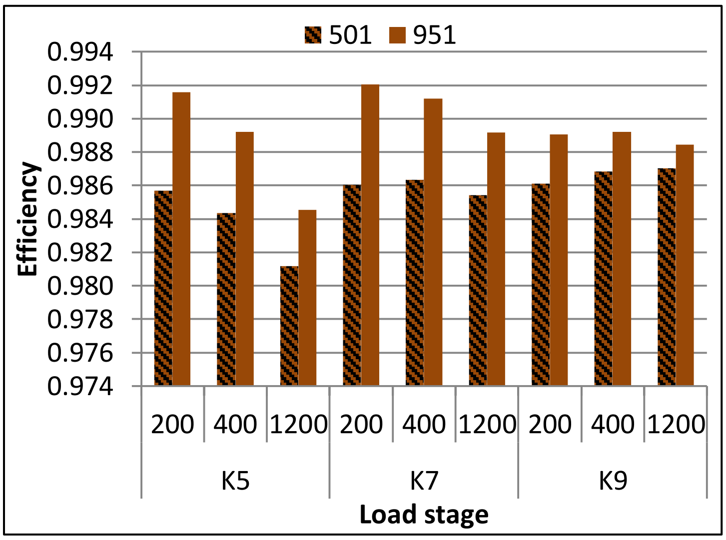

Figure 78: Efficiency comparison of 501 and 951 gear geometries using MINR oil.

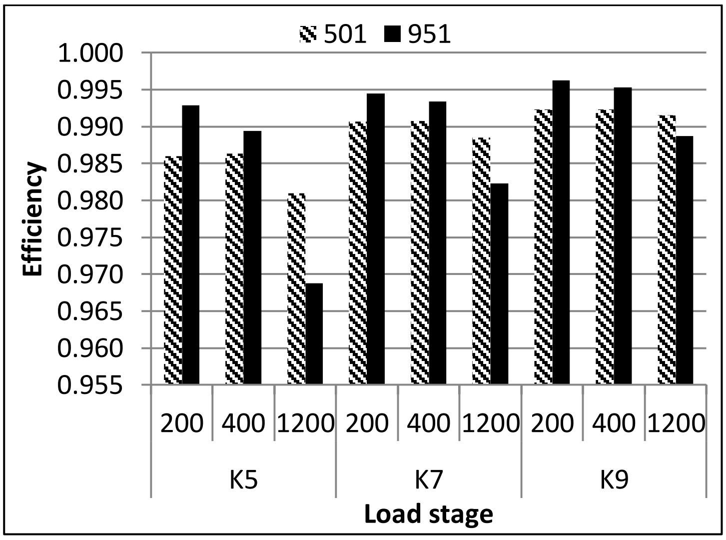

Figure 79: Efficiency comparison of 501 and 951 gear geometries using PAGD oil.

5 Gearbox Power loss

5.1 Introduction

Inside a gearbox different sources of power loss are generated. These losses can be divided in load dependent and no-load dependent losses [6].

The load losses are generated by the contact between surface of the power transmitting components (bearings and gears). The no-load losses are generated by the motion of the parts inside the oil, or the relative motion between surfaces separated by a film and occurs in gears, rolling bearings and seals.

- The lubricant viscosity and density, the immersion depth of the mechanical parts (for bath lubrication), the operating conditions and the internal housing design are variables that influence the no load losses. The load losses are affected by the coefficient of friction, the sliding and rolling velocity and the transmitted load and the contact geometry.
- To define the coefficient of friction it is necessary to know the lubrication regime, since the coefficient of friction is dependent on the film thickness. The lubrication regime can be divided into [1] (see Figure 80):
- **Boundary film lubrication ($\Lambda<0.7$):** absence of the hydrodynamic lubricant film between the surface thus allowing metal to metal contact. The normal load is supported by the roughness tips;
- **Mixed film lubrication ($0.7<\Lambda<2.0$):** partial formation of the lubricant film and the load is supported both by the roughness tips and the EHD lubricant film;
- **Full film lubrication ($\Lambda>2.0$):** the surfaces are totally separated by the EHD lubricant film preventing the metal-metal contact.

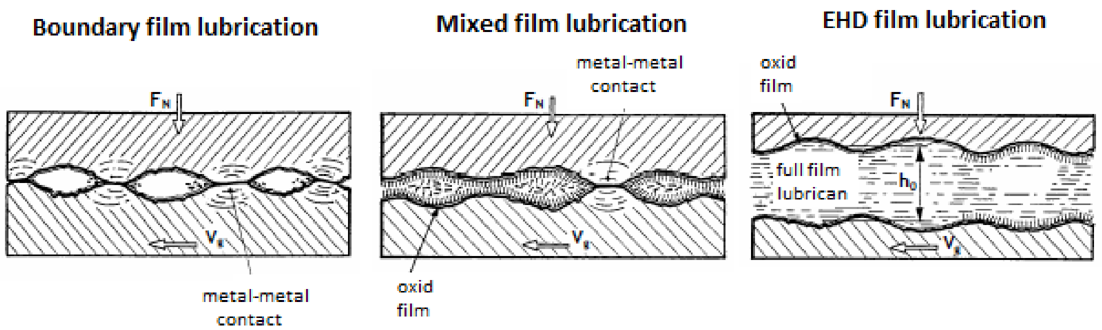

Figure 80: Lubrication regimes in EHD lubrication

The total Power loss P_V is given by the sum of different sources within a gearbox. On this work the losses were divided according equation (0.1) where P_{VZ0} and P_{VD} are load independent losses of gears and seals respectively, P_{VZP} and P_{VL} are the load losses of gears and bearings, respectively and P_{VX} is the auxiliary power loss due to unknown sources.

$$P_V = \sum P_{VZ0} + \sum P_{VZP} + \sum P_{VL} + \sum P_{VD} + \sum P_{VX} \text{ [6]} \tag{0.1}$$

All different sources of power losses will be detailadelly presented in the next paragraphs.

5.2 No load dependent losses

5.2.1 Seals power losses

In most case the seals losses are less than 0.01% of the nominal transmitted power, a very small value when compared to the other losses in the gearbox [7], but this is only valid for low tangential speed.

Simrit® presents equation 0.2 to calculate the seal losses.

$$P_{VD} = 7.69 \times 10^{-6} \cdot {d_{sh}}^2 \cdot n \tag{0.2}$$

Where n is the rotational speed [rpm] and d_{sh} the inner diameter [mm].

5.2.2 Gear spin power losses

Gear spin power losses are directly related to the lubrication method [8]. For dip lubrication the components are partially submerged and the losses are designated by churning losses and can be obtained by:

- the interactions of each of the individual gears with the surrounding medium;
- the interaction of the gear pair at the gear mesh interface, with squeezing and pocketing being the dominant modes of power losses.

In case that the lubrication type is jet lubrication the power losses are due to the interaction between air or air-oil mixture and the rotating gears and are designated by windage losses. The windage losses for gears are:

- Losses due to the squeezing of compressible air/oil-air mixture through the mating contacts;
- Drag power losses due to air drag along the teeth and sides of the rotating gear.

The windage power losses can be the dominant losses in dip lubrication when oil level is very low [9].

5.2.2.1.1 Windage power losses

The power loss tests were performed in jet lubrication. To calculate the total windage loss, the model of Mauz (1987), Maurer (1994) and Ariura (1974) were used. This model combines three different terms that are the squeeze loss $P_{squeeze}$, Ventilation loss P_{vent} and impulse loss P_{impuls}.

$$P_{windage} = P_{squeeze} + P_{vent} + P_{impuls} \tag{0.3}$$

Squeeze loss is dependent on the oil volume flow and nozzle-position (see Figure 81) [8]. Mauz presents equation 0.4 to calculate it.

$$P_{squeeze} = \overline{\omega} \cdot 4.12 \cdot C_1 \cdot \rho \cdot \dot{Q}_e{}^{0.75} \cdot r \left(v_t \cdot b \cdot m \cdot \frac{\upsilon}{\upsilon_0} \right)^{0.25} \cdot \left(\frac{h_t}{h_{t0}} \right) \tag{0.4}$$

Where $\dot{Q}$ is the oil volume flow, C_1 the scaling factor for gravity effect (1 for UC and DCC and 0.9 for DC, UCC), ν_0 the reference kinematic viscosity [cst], ν the kinematic viscosity [cst] at oil operating temperature, v_t the rotational speed of the gear, h_t the tooth height and h_{t0} the reference tooth height.

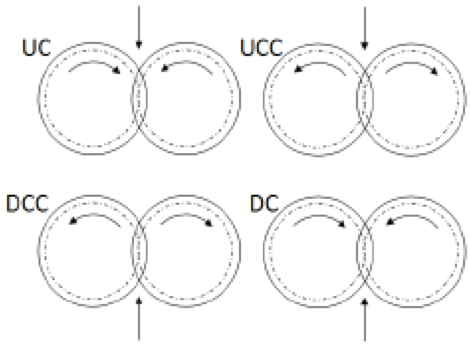

Figure 81: different case of jet lubrication

Table 22 displays the coefficient to account for gravity for each lubrication type.

Table 22: Coefficient to account for gravity.

Lubrication case	Coefficient to account for gravity
UC	1
UCC	1
DCC	0.9
DC	0.9

The impulse loss is due to the oil acceleration by the gears and is dependent on the oil stream velocity, oil flow and gear operating speed, it can be calculates by equation 0.5 according to Ariura [8].

$$P_{impulse} = \overline{\omega} \cdot C_x \cdot \rho \cdot \dot{Q}_e \cdot r \cdot (|v_t| - v_s) \tag{0.5}$$

Where C_x is the coefficient to account for gravity (Table 22), v_t the rotational speed of the gear and v_s the velocity of the oil stream.

The ventilation loss torque is composed by two different components that are the loss of the individual gear $P_{vent,id}$ and the loss in the mating zone $P_{vent,ca}$. Maurer present equations 0.6, 0.7 and 0.8 to determine the ventilation losses.

$$P_{vent} = P_{vent,id} + P_{vent,ca} \tag{0.6}$$

$$P_{vent,id} = 1.37 \times 10^{-9} \cdot {v_t}^{1.9} \cdot d^{1.5} \cdot b^{0.52} \cdot m^{0.69} \cdot \lambda_{wand} \tag{0.7}$$

$$P_{vent,ca} = 1.17 \times 10^{-6} \cdot {v_t}^{1.95} \cdot d^{1.5} \cdot i^{0.73} \cdot b^{1.37} \cdot m^{0.69} \cdot \lambda_{wand} \tag{0.8}$$

To calculate the churning losses, on this work, the tests were performed at load stage k1 that is a stage representative of no-load losses. The churning losses for each speed are calculated subtracting the seals and bearings losses to the experimentally measured power losses, in load stage k1, as is show in equation (0.9).

$$P_{VZP0} = P_{Lexp} - P_{VL} - P_{VD} \tag{0.9}$$

5.3 Load dependent losses

5.3.1 Rolling Bearing power loss

Power losses in bearings are generated due to friction between the different parts of the bearings. These are dependent on the applied load in the bearing, rotational speed, size and type of rolling bearing, the quantity of lubricant and the lubricant properties [10].

In order to calculate the total friction generated in rolling bearings, SKF presented recently a new model [10]. This model considers separately four different friction sources in rolling bearings as show in equation (0.10).

$$M = \varphi_{ish}\varphi_{rs}M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{0.10}$$

Where:

- M – Total frictional torque, Nmm;
- M_{rr} – rolling frictional torque, Nmm;
- M_{sl} – sliding frictional torque, Nmm;
- M_{seal} – frictional torque of the seal(s), Nmm;
- M_{drag} – frictional torque of drag losses, churning, splashing etc., Nmm;

- φ_{ish} – inlet shear heating reduction factor;
- φ_{rs} – kinematic replenishment/starvation reduction factor.

The total bearing power loss is given by:

$$P_{VL} = M \times n \times \frac{\pi}{30} \times 10^{-3} \quad (0.11)$$

Table 23 shows all model parameters necessaries to calculate rolling bearings power loss.

Table 23: QJ 308 N2MA and NJMA 406 bearings parameters.

Parameter	QJ 308 N2MA	NJ 406
R1	4.78E-7	1.09E-6
R2	2.42	0
R3	1.40 E-12	0
S1	0.012	0.16
S2	0.9	0.0015
S3	1.40E-12	0
Krs	0.00000003	0.00000003
Kz	3.1	5.1
Ks1	0	0
Ks2	0	0
Beta	0	0
KL	0	0.65
irw	1	0

For further information on the skf model, please consult [10], [11]and in [12].

5.3.2 Gear mesh power loss

The mesh power losses are dependent on the transmitted power (P_a), the mean value of the coefficient of friction in the gear contact (μ_{mz}) and the gear loss factor (H_v) [7]. Höhn et al [7], present equation 0.12 to calculate gear mesh losses:

$$P_{VZP} = P_a \cdot \mu_{mz} \cdot H_V \quad (0.12)$$

Where P_{VZP} is the mesh power loss, P_a is the transmitted power, μ_{mz} is the mean coefficient of gear friction and H_v is the gear loss factor.

The gear loss factor (H_V) is calculated using equation 0.13.

$$H_V = \frac{\pi \cdot (u+1)}{z_1 \cdot u \cdot \cos \beta_b} \cdot (1 - \varepsilon_\alpha + {\varepsilon_1}^2 + {\varepsilon_2}^2) \quad (0.13)$$

ε_α is the transverse contact ratio, ε_1 and ε_2 are the pinion and gear addendum contact ratio, respectively.

The mean value of the coefficient of friction is very difficult to determine. The coefficient of friction between gear teeth is one of the most important variables since it is directly linked to the contact temperature, the probability of failure occurrence in the efficiency of the gear [1].

Höhn et al after numerous tests in twin disk machine and gear test rigs, present an empirical equation for the mean coefficient of friction between gear tooth [7]:

$$\mu_{mz} = 0.048 \cdot \left(\frac{\frac{F_{bt}}{b}}{V_{\Sigma C} \cdot \rho_C} \right)^{0.2} \cdot \eta_{oil}{}^{-0.05} \cdot R_a{}^{0.25} \cdot X_L \tag{0.14}$$

Where F_{bt} is the tangential force [N], b is the face Width [mm], $V_{\Sigma C}$ is the rolling speed [m/s], ρ_c is the curvature at pitch radius [mm], R_a is the roughness [µm] and X_L the lubricant factor.

In this work case, the meshing losses are calculated subtracting the seals power losses, the bearings power losses and the churning losses to Experimental power losses, as is shown in equation (0.15).

$$P_{VZP} = P_{exp} - P_{VL} - P_{VD} - P_{VZP0} \tag{0.15}$$

The coefficient of friction is calculated by the division between the meshing losses and the product of the input power by gear losses factor (see equation (0.16)).

$$\mu_{mz} = \frac{P_{VZP}}{P_{in} \cdot H_V} \tag{0.16}$$

5.4 Model Validation

The numerical model was implemented and run in *MATLAB®*. The numerical model aims to calculate the different sources of power loss occurring in FZG test machine.

In order to validate the numerical model and identify the different sources of power loss will be presented and treated, in more detail, the different sources of power loss of the oils that have the best and the worst behaviour, PAGD and MINR respectively. The results of the numerical model are presented for each oil in appendix A.2.

5.4.1 C40 gear design

The results will be divided for the two oils selected to identify their influence on the power losses, as is shown in Figure 82 to Figure 84.

In Figure 82 is represented the variation of the churning losses with the rotational speed. Despite the small difference at range speed 200 to 400 MINR present less churning losses than PAGD. For higher speed (1200) PAGD distinguished with less churning loss despite its higher viscosity.

The meshing losses (Figure 83) at k1 load stage are approximately zero because k1 load stage is representative of the no-load losses. When the load and the speed increases the meshing losses increase and MINR present higher losses than PAGD, such difference can be explained due to the method used to determine the churning losses.

At 200 rpm the rolling bearing losses (Figure 84) are approximately the same. Increasing the speed (range speed 400 to 1200) the difference between the bearings losses increases with the speed and the load. The PAGD have higher losses due to the rolling torque losses due to the higher viscosity.

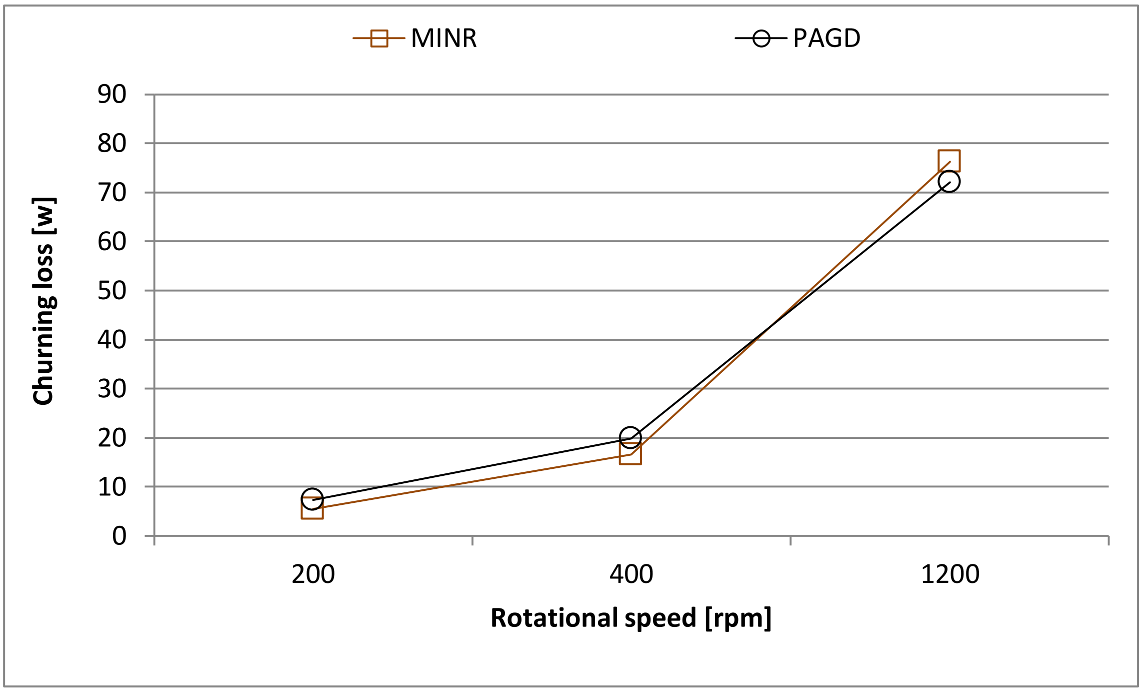

Figure 82: C40 gear design Churning losses.

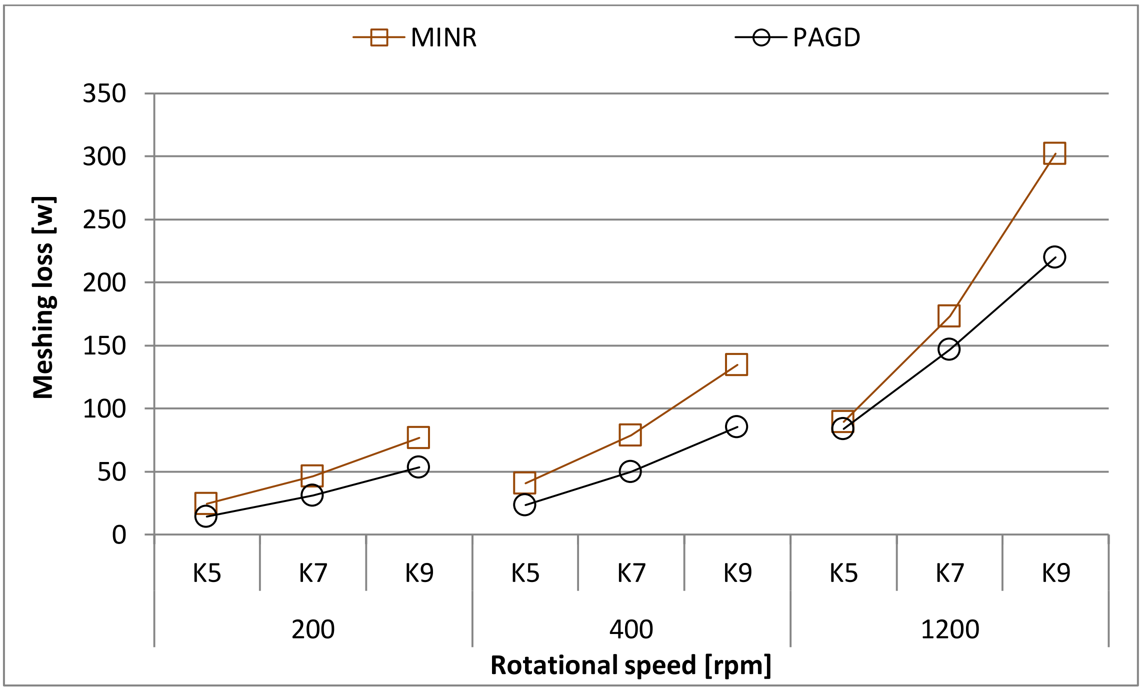

Figure 83: C40 gear design Meshing losses.

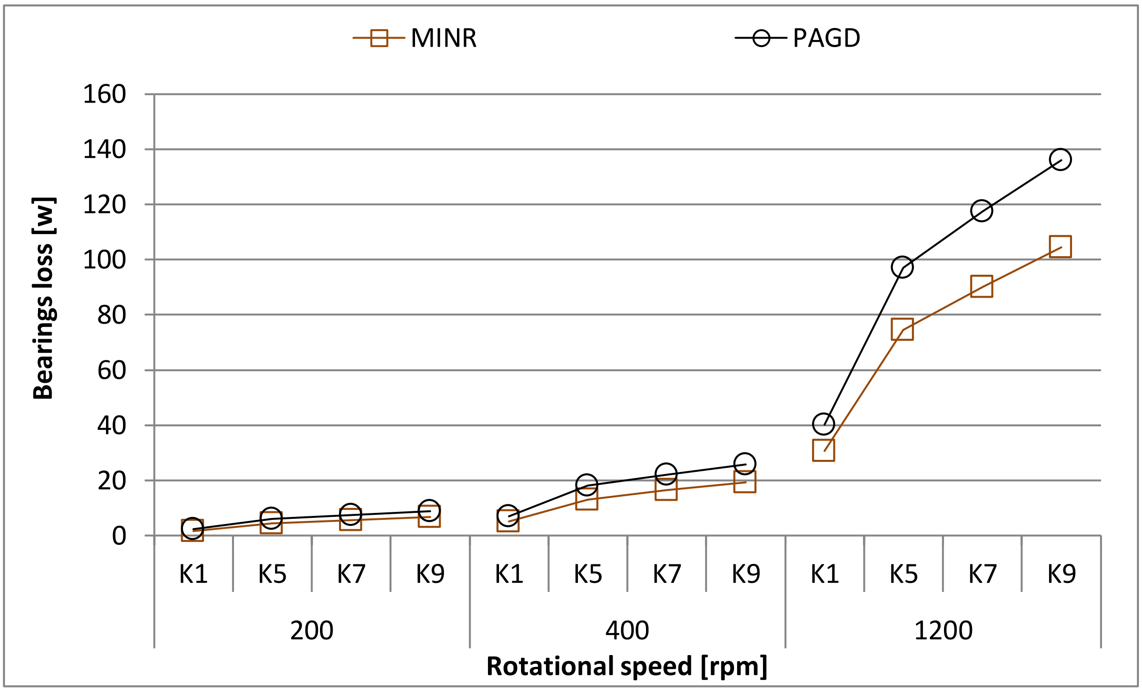

Figure 84: FZG gearboxes Bearings losses.

5.4.2 501 gear design

Such as for C40 gear design, it will be presented and discussed the results of each source of the test gearbox with a helical gear 501 lubricated with MINR and PAGD oils.

In some cases the churning losses, k1 load stage, is very higher. This high value can be due to an auxiliary loss that occur in some oils (see appendix A.2), that promotes an increase temperature of the oil after passing the contact.

In Figure 85 to Figure 87 it is represented separately the different sources of power loss existent in Test gearbox.

In Figure 85, churning losses are presented, the PAGD present higher losses than MINR for all speed range. The difference between PAGD and MINR losses increases with the speed increase.

In Figure 86, Meshing losses representative, and for all range load the MINR present high losses than PAGD independently of the speed. With speed increase the difference between MINR and PAGD losses increase achieving a difference 4 times higher, approximately.

The bearing losses (Figure 87) at range speed 200 to 400 rpm are approximately the same. For high speeds (1200 rpm) the PAGD distinguishes itself with higher losses, independently of the load stage.

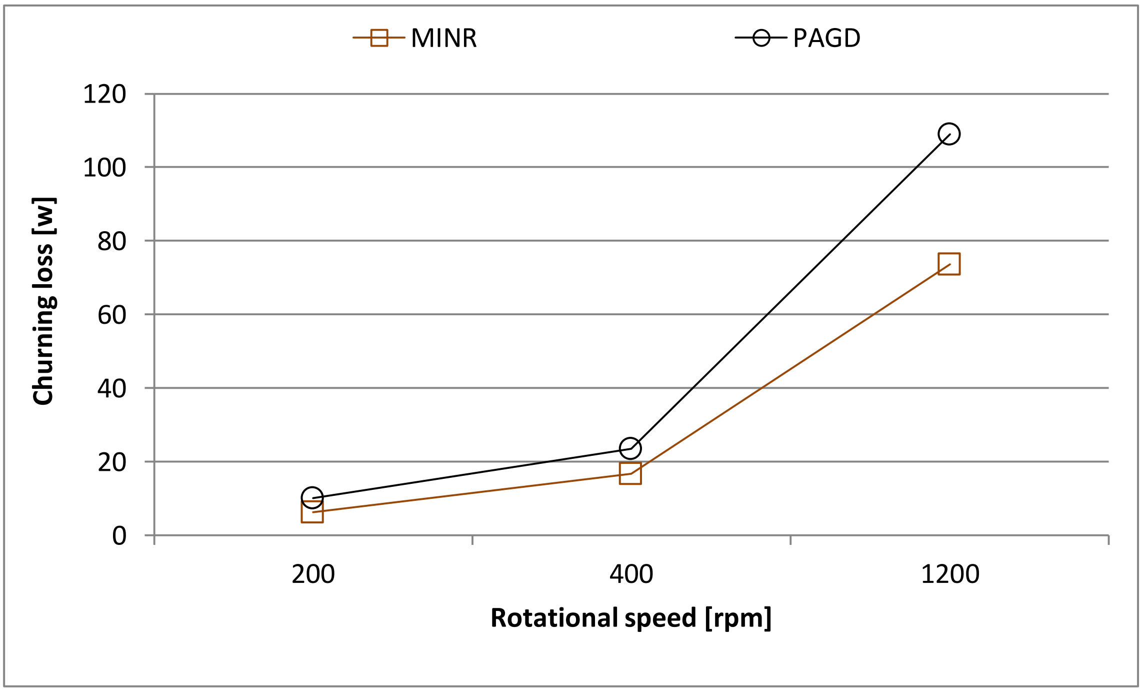

Figure 85: 501 gear design Churning losses.

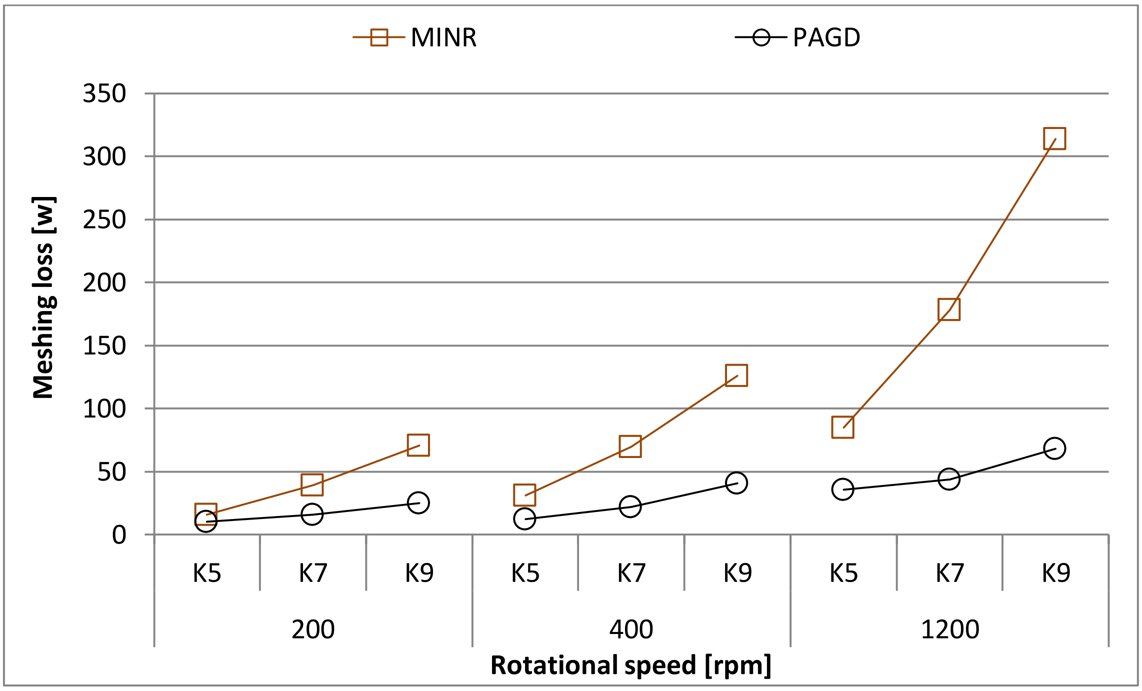

Figure 86: 501 gear design Meshing losses.

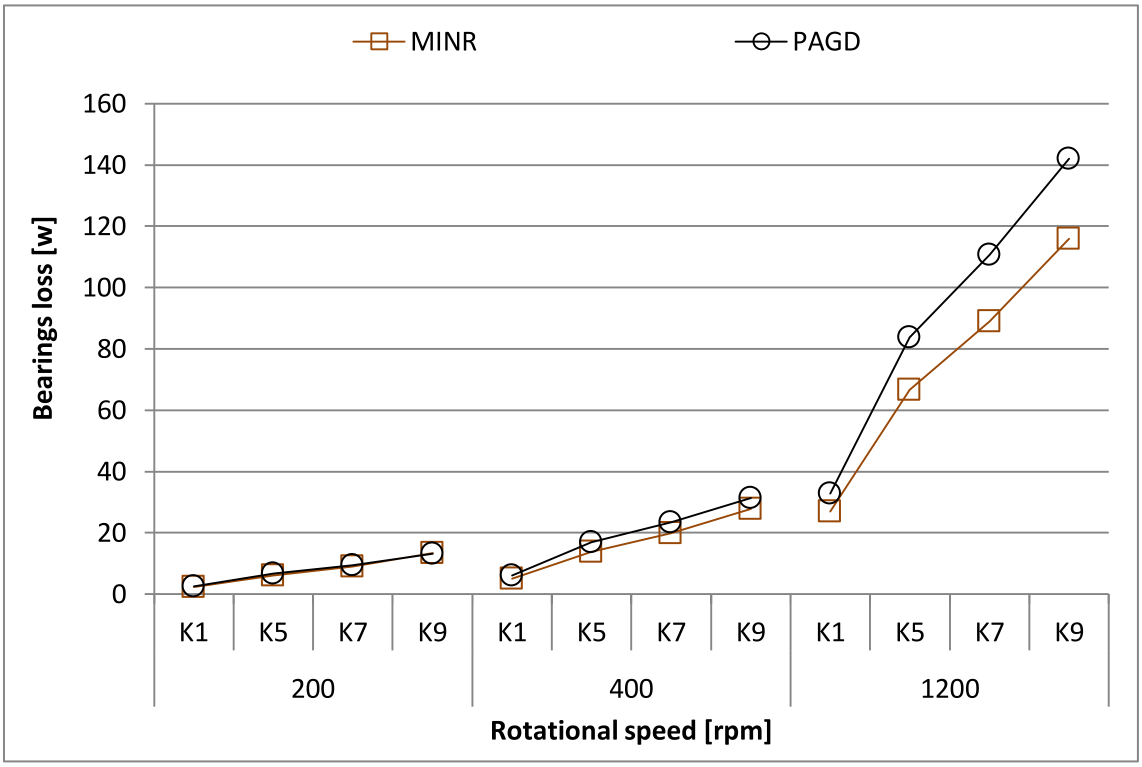

Figure 87: FZG test gearbox bearings losses.

5.4.3 951 gear design

Such as for C40 and 501 gear design, it will be represented and discussed the results of each source of the test gearbox with a helical gear 951 lubricated with MINR and PAGD oils.

In order evaluate the different sources of power loss for each oil type, Figure 88 to Figure 90 present the different sources of power loss separately.

The experimental results of the oils torque loss at k1 load stage with 951 is higher than at load stage k5 and in some cases higher than k7 load stage (see appendix A.2). This high value is due of the auxiliary losses present in load stage k1. In 951 gear design will be used the 501 gear design churning losses value see Figure 88, and the difference for the value measured is the auxiliary loss.

In Figure 89, that represents the meshing losses of MINR and PAGD have approximately the same value. Despite the smaller difference and, at higher loads, PAGD presents the lowest value of meshing losses.

The oil that promotes higher losses in the bearings (Figure 90) is the PAGD for all load stage and speed range. At 200 rpm the bearings is approximately the same.

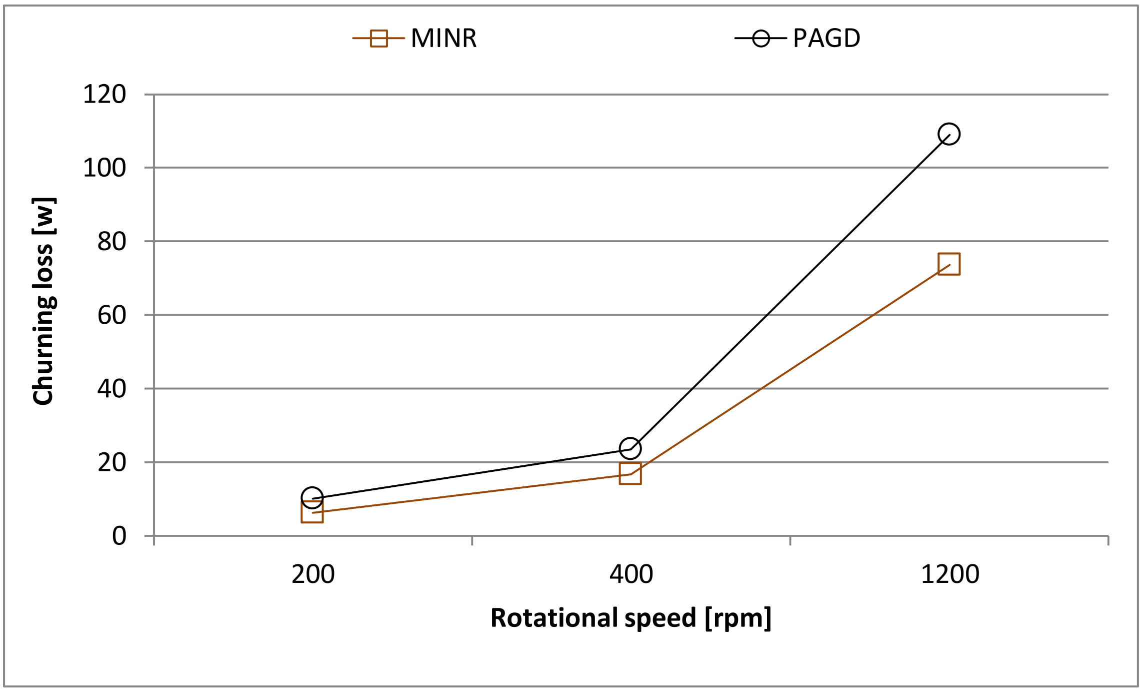

Figure 88: 951 gear design Churning losses.

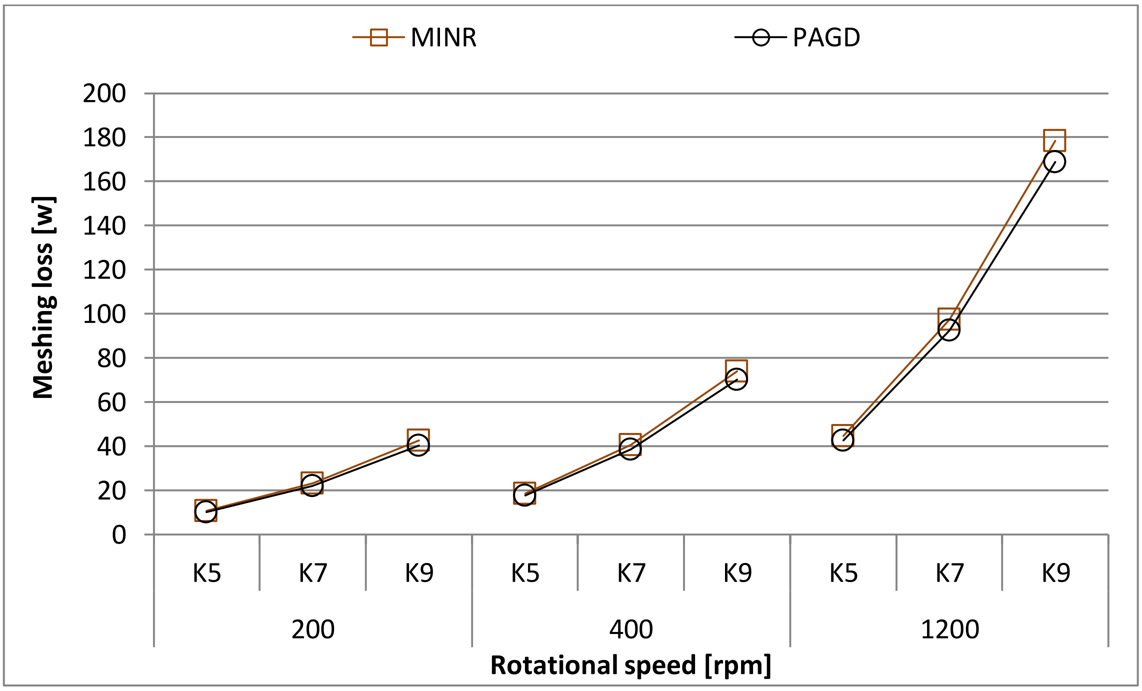

Figure 89: 951 gear design Meshing losses.

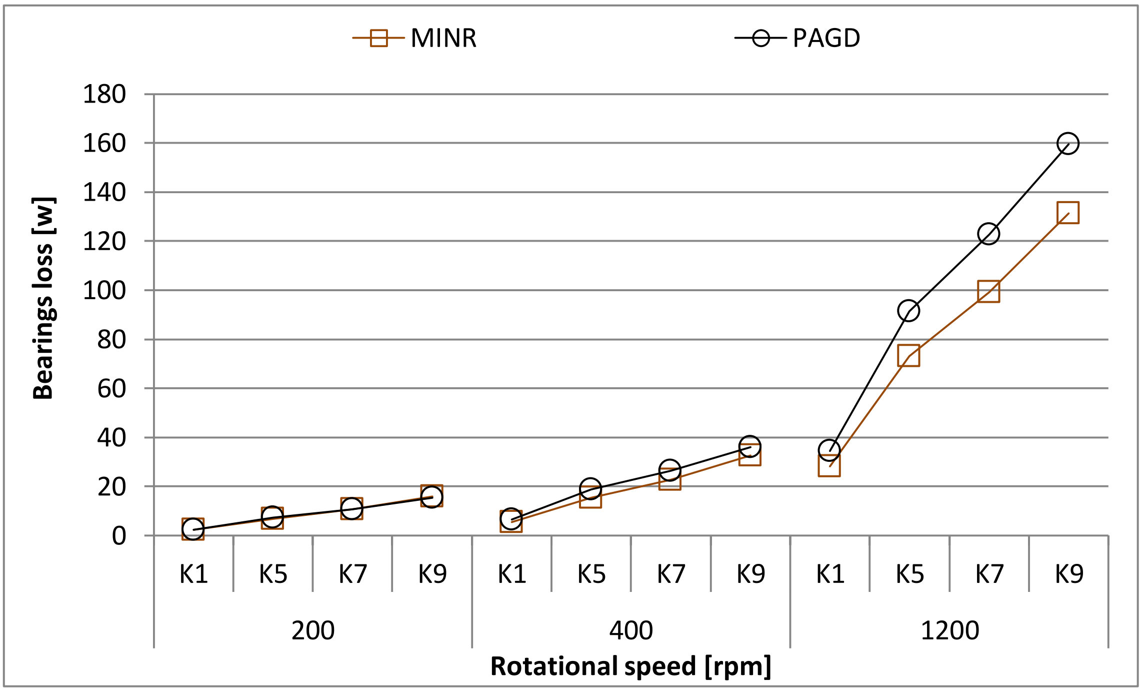

Figure 90: FZG test gearbox bearings losses.

5.4.4 C40 vs 501 vs 951

In this section will be compared the different losses for each gear design tested. Besides the comparison between gear geometries will also be done the comparison between two oils, PAGD and MINR.

As stated in section 5.4.3 the churning losses value used in 951 gear design is equal to the 501 gear design.

The C40 and the 951 gear design churning losses is approximately the same when lubricated with MINR (see Figure 92). In the case that the gears are lubricated with PAGD (Figure 91) C40 gear design present lower losses than 501 gear design. The difference between the gears losses, increase when speed increase.

The gears design that present lower meshing loss (Figure 93 and Figure 94) varies with the oil. For the case that the gears are lubricated with MINR the 951 gear present lower losses. The C40 gear design and 501 gear design has very similar value, presenting the higher value of Meshing losses. For the case that are lubricated with PAGD and 501 gear design presented lower losses and C40 gear design present higher losses for all speed range. Both for PAGD and MINR, the differences between the gears meshing losses increase with the load.

The rolling bearing losses, Figure 95 and Figure 96, both MINR and PAGD has the same behavior, approximately. At low speed both C40, 501 and 951 gear design present the same losses. At 400 and 1200 rpm the 951 gear design present higher losses and 501 gear design present lower losses being more notorious at k7 and k9 load stage. The difference between gear design losses increase when load increase. At 400 rpm the bearing losses with 501 gear design and 951 gear design present approximately the same losses.

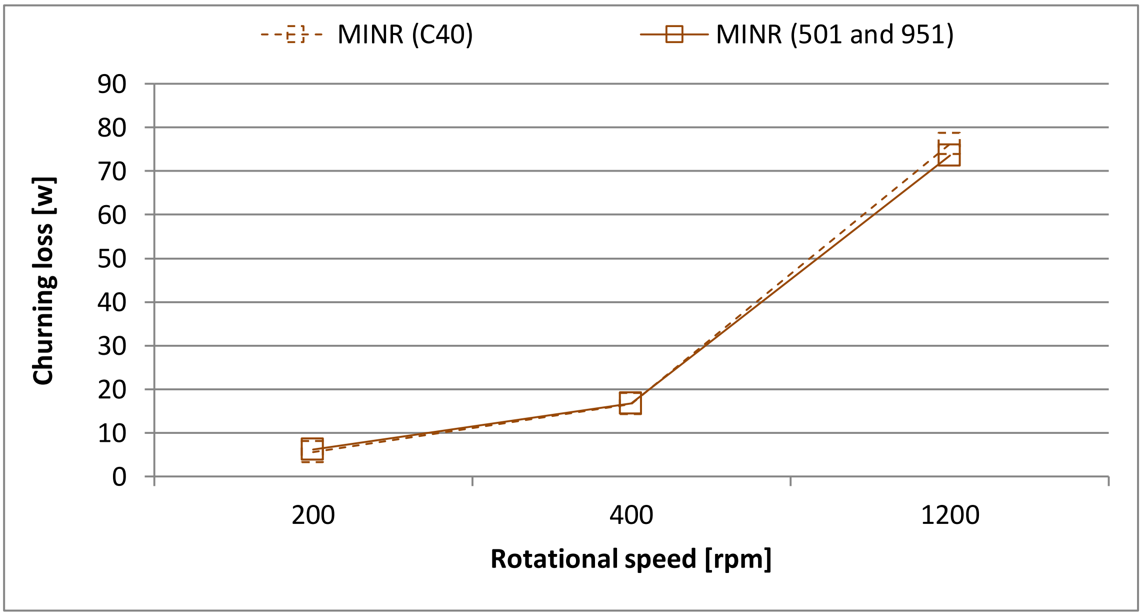

Figure 91: C40 vs 501 vs 951 Churning losses lubricated with MINR.

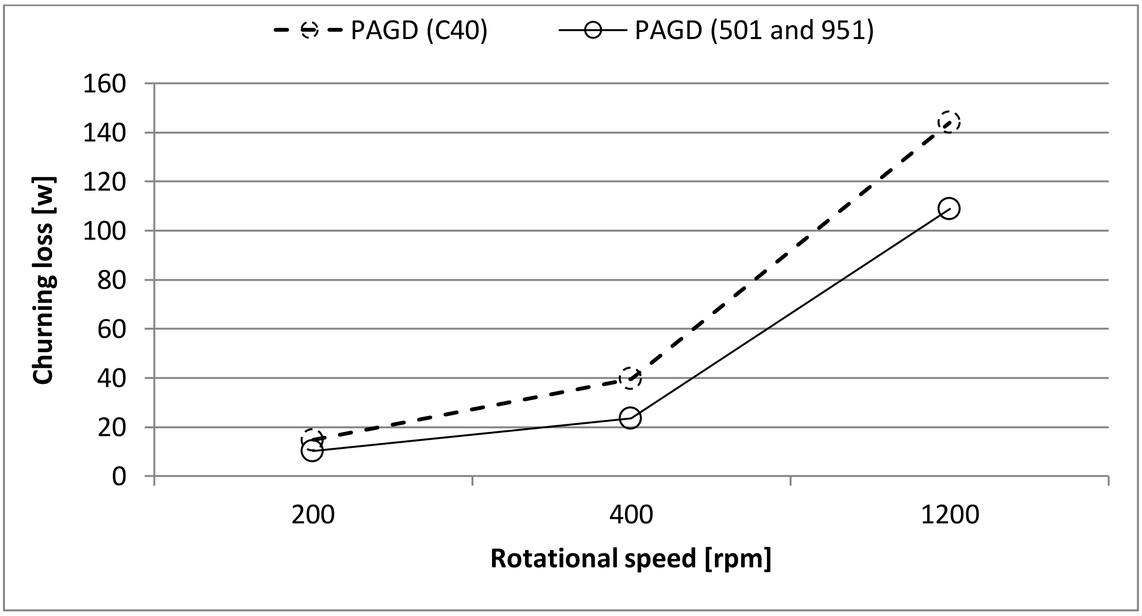

Figure 92: C40 vs 501 vs 951 Churning losses lubricated with PAGD.

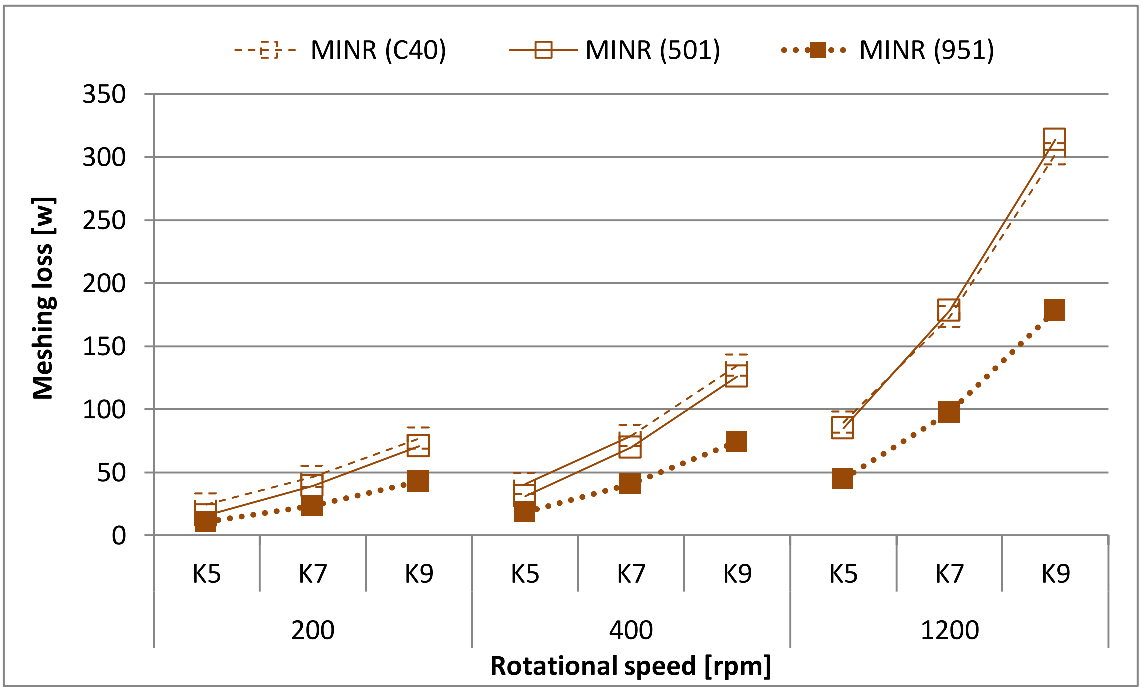

Figure 93: C40 vs 501 vs 951 Meshing losses lubricated with MINR.

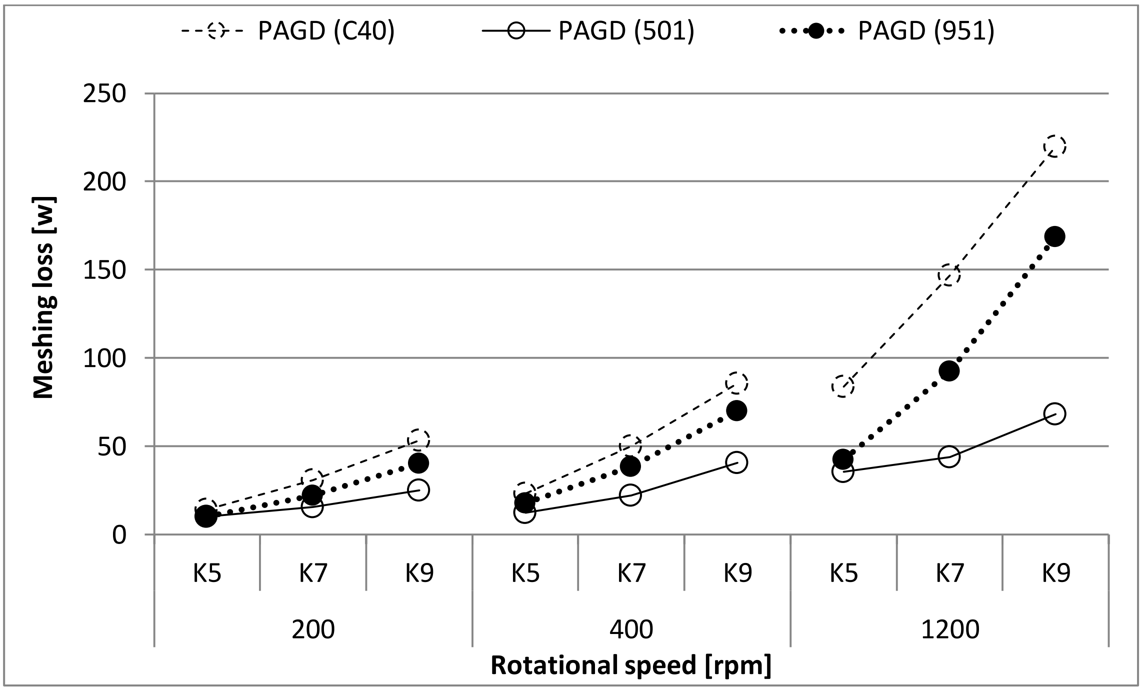

Figure 94: C40 vs 501 vs 951 Meshing losses lubricated with PAGD.

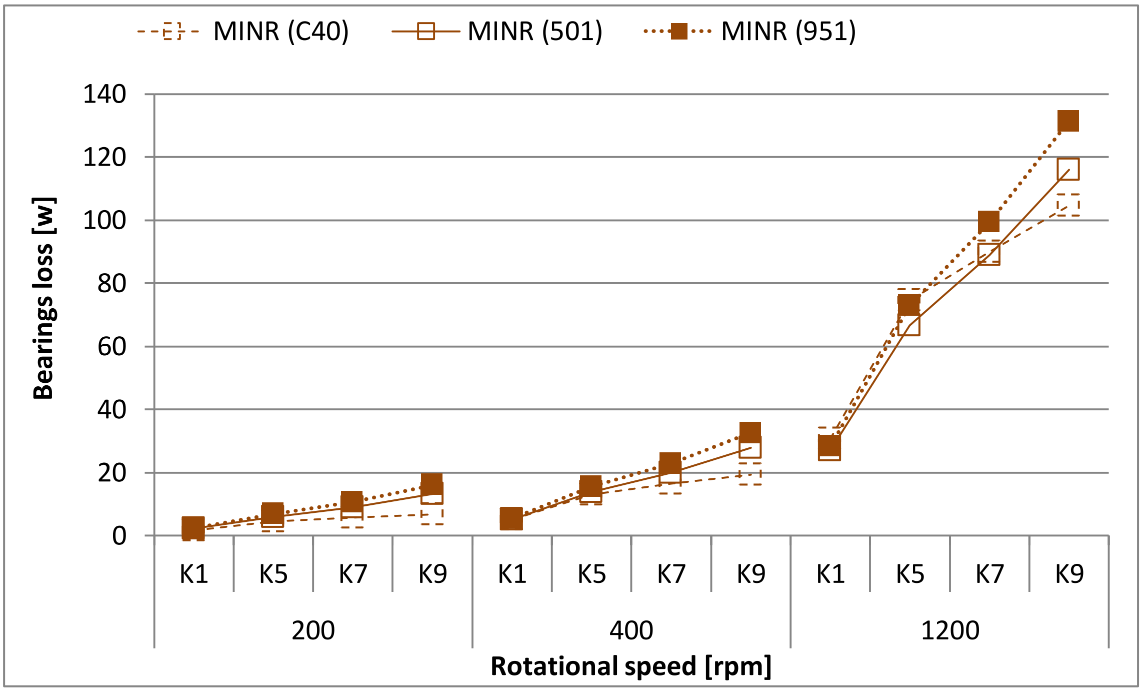

Figure 95: C40 vs 501 vs 951 Bearings losses lubricated with MINR.

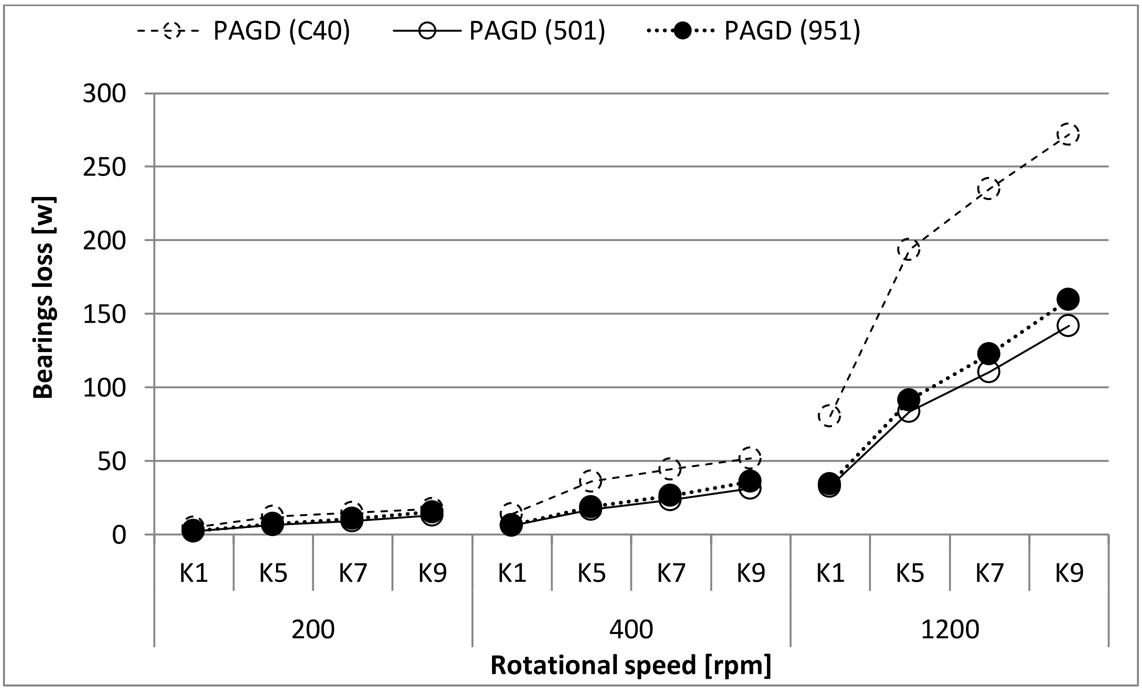

Figure 96: C40 vs 501 vs 951 Bearings losses lubricated with PAGD.

5.4.5 Coefficient of friction

The coefficient of friction is calculated by equation (0.16). In Figure 97 to Figure 99 is presented the coefficient of friction for each gear design.

Looking to Figure 97, which represent C40 gear design coefficient of friction, is visible that MINR present higher coefficient of friction than PAGD for all test conditions. At higher speed (1200) rpm and at load stage k5 MINR and PAGD present a most similar coefficient of friction. It is also noteworthy that at 1200 rpm the C40 gear design coefficient of friction when lubricated with MINR increase with the load increase and when lubricated with PAGD decrease with the load increase.

The coefficient of friction by 501 gear design (Figure 98) is higher when lubricated with MINR that when lubricated with PAGD for all test conditions. At all speed range the coefficient of friction of MINR increase with the load increase and the coefficient of friction of PAGD decrease with the load increase.

In Figure 99, 951 gear design coefficient of friction representative, is visible that the coefficient of friction is higher when lubricate with MINR oil than when lubricate with PAGD for all test conditions. The difference between MINR and PAGD is approximately the same for all test conditions.

In Figure 100 and Figure 101 is made the coefficient off friction comparison between the tested gear design lubricated with MINR and PAGD, respectively.

Comparing the coefficient of friction of the tested gears design lubricated with MINR (Figure 100), is visible that at 200 rpm 501 gear design and C40 gear design present the lowest and the highest, respectively, coefficient of friction for k5 load stage. With the load increase, k7 and k9 load stage, the C40 don´t has variation presenting the lowest value and 951 present the highest coefficient of friction. At 400 rpm and at k5 load stage the coefficient of friction is approximately the same for all tested gears. At k7 and k9 load stage 501 and 951 gear design presented the highest coefficient of friction. For highest speed C40 gear design and 501 gear design present the lowest and the highest coefficient of friction, respectively for all load stage. Also it is noteworthy that 501 and 951 gear design has a small difference, decrease with the load increase.

Looking to Figure 101, that represent the coefficient of friction of the tested gears design lubricated with PAGD, 951 gear design and 501 gear design present the highest and the lowest coefficient of friction, respectively, for all tested conditions. The 501 gear design at 200 rpm and 1200 rpm and the C40 gear design at 1200 rpm, the coefficient of friction decrease with the load increase. At 400 rpm the 501 gear design coefficient of friction has a small variation with the load increase. The 951 gear design at all tested speed and the C40 gear design at 200 and 400 rpm, the coefficient of friction increase with the load increase.

Figure 102 and Figure 103 represent the variation of the ratio between meshing power loss and the input power with the speed increase.

When the tested gears are lubricated with MINR (Figure 102) and at 200 rpm and 400 rpm, C40 gear design has higher ratio than the others gears design, independently load stage, and so higher meshing losses. At 1200 rpm and at k5 load stage C40 design also has higher ratio but, at k7 and k9 load stage, the 501 gear design present the higher ratio. The lowest ratio is present by 951 gear design, for all test conditions, and so the lower meshing losses value, as was shown in Figure 93.

In the case that the tested gears are lubricate with PAGD oil (Figure 103) C40 and 951 gear design present the higher and lower ratio, respectively, for all test conditions. Also it is noteworthy that at 200 rpm and at k5 load stage 501 and 951 has approximately the same value.

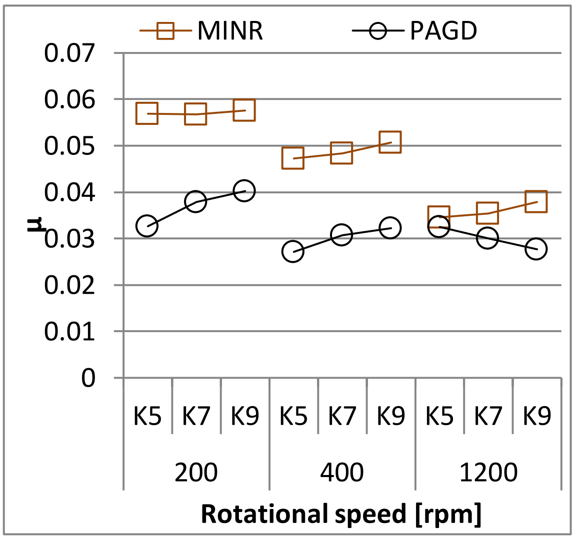

Figure 97: C40 gear design coefficient of friction.

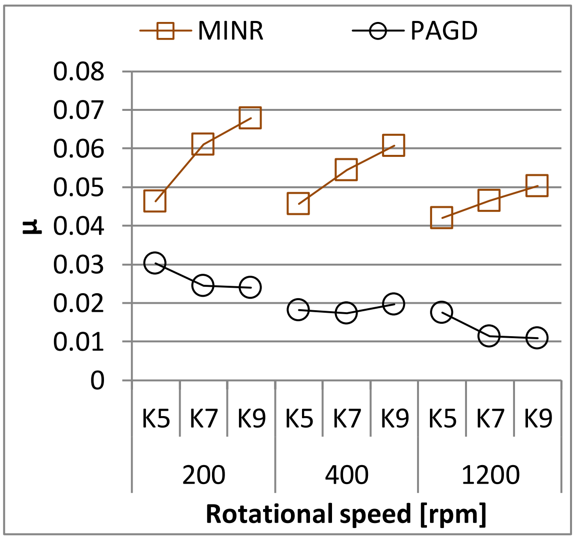

Figure 98: 501 gear design coefficient of friction.

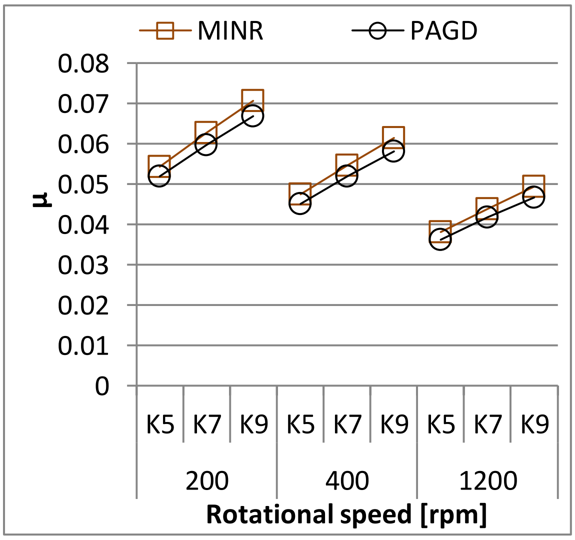

Figure 99: 951 gear design coefficient of friction.

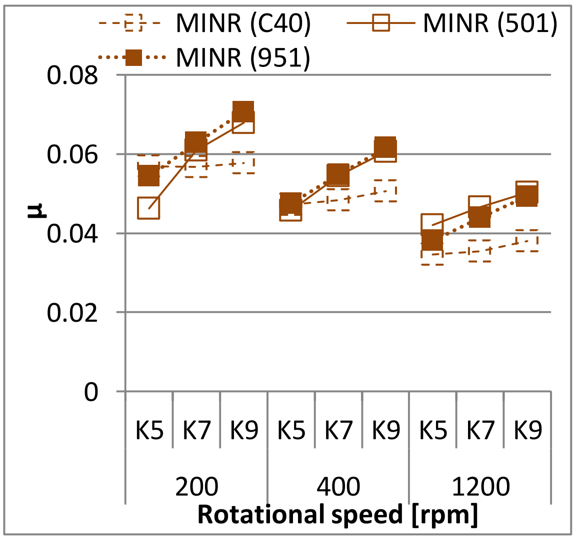

Figure 100:MINR oil coefficient of friction.

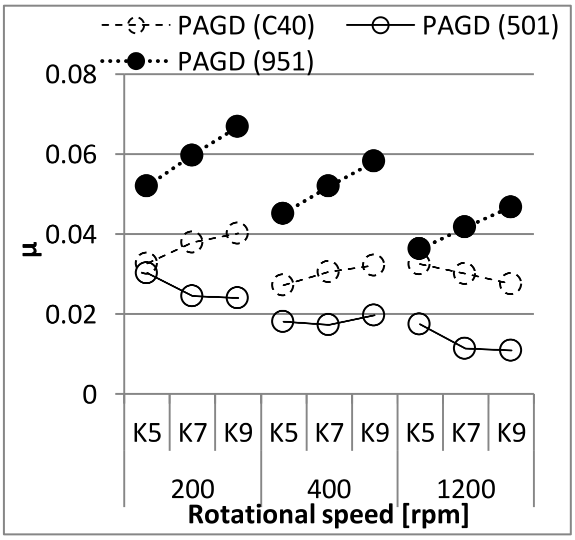

Figure 101: PAGD oil coefficient of friction.

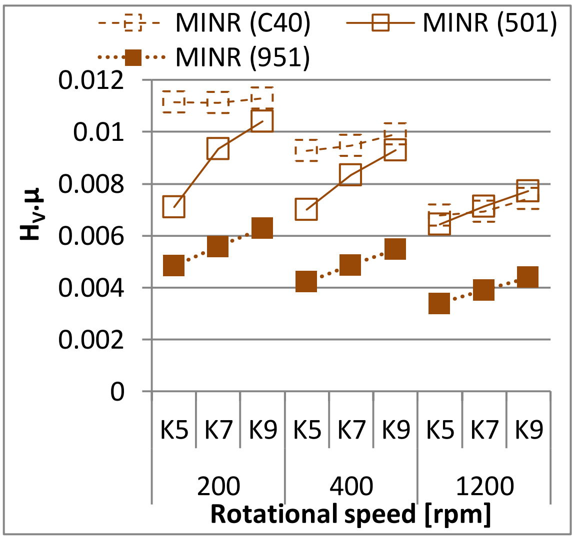

Figure 102:MINR oil P_{VZP}/P_{in}.

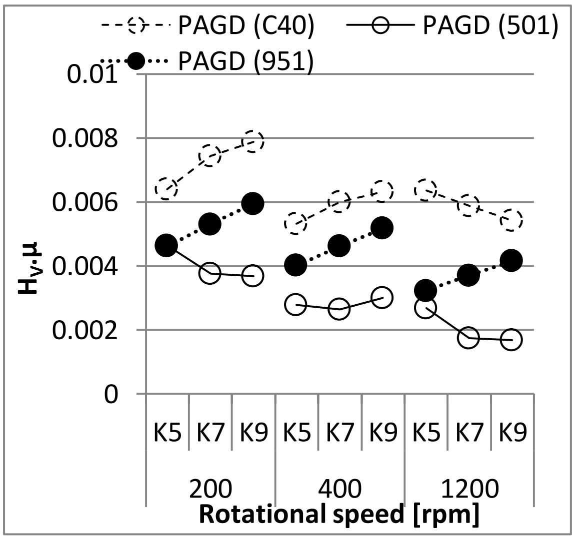

Figure 103: PAGD oil P_{VZP}/P_{in}.

5.4.6 Model validation for load stage K11

In order to validate the numerical model will be used the experimental results of the test performed at k11 load stage. Must be noted, that for k11 load stage, the power loss of the drive gearbox was not measured separately.

As first step, it was calculated the theoretical coefficient of friction for the different test conditions through the equation (5.14), considering the oil properties and the gears average roughness.

$$\mu_{mz} = 0.048 \cdot \left(\frac{\frac{F_{bt}}{b}}{V_{\Sigma C} \cdot \rho_C} \right)^{0.2} \cdot \eta_{oil}{}^{-0.05} \cdot R_a{}^{0.25} \cdot \tag{0.17}$$

The lubricant factor $X_{L,}$ for each test conditions that adjust the equation (5.6) for each oil type, can be calculated by:

$$\boldsymbol{X_L} = \frac{\mu}{\mu_{Michaelis}} \tag{5.18}$$

Having the lubricant factor for each test conditions, an average of all X_L values was determined and thus calculated the theoretical coefficient of friction for load stage k11. With the K11 coefficient of friction, the Meshing losses were determined by:

$$P_{VZP} = P_{in} \cdot \mu_{mz} \cdot H_V \tag{0.19}$$

In order to obtain the total power loss will be added the other sources of power loss (P_{VL}, P_{VD} and P_{VZP0}). This procedure will be made for the oils MINR and PAGD. The results for MINR oil are presented in Table 24and Table 25 and for PAGD oil are presented in Table 26 and Table 27.

As shown in Table 25, the MINR torque loss calculates by numerical model has a maximum error of the 2.3% at 200 rpm and for the other speeds the error is below one. With the obtain error it can be concluded that the numerical model works perfectly for the MINR. The higher error for 200 rpm can be addressed for the boundary lubrication regime verified on C40 gear design for this load.

The calculate torque loss by the numerical model for the PAGD has a maximum error of 12% at 200 rpm. The errors at the other speeds are approximately 6% and 8% being an acceptable error.

Table 24: Experimental coefficient of friction, theoretical coefficient of friction and lubricant factor for MINR oil.

Speed	Load stage	μ		$\mu_{Michaelis}$		X_L	
		501	C40	501	C40	501	C40
200	K5	0.046	0.057	0.050	0.060	0.955	0.947
	K7	0.061	0.057	0.056	0.068	1.107	0.833
	K9	0.068	0.058	0.062	0.075	1.118	0.767
400	K5	0.046	0.047	0.043	0.052	1.082	0.905
	K7	0.054	0.048	0.049	0.059	1.135	0.815
	K9	0.061	0.051	0.054	0.065	1.148	0.774
1200	K5	0.042	0.035	0.035	0.042	1.241	0.824
	K7	0.047	0.035	0.039	0.048	1.210	0.743
	K9	0.050	0.038	0.043	0.053	1.187	0.722
Average						1.131	0.814

Table 25: MINR power loss sources at k11 load stage.

	PL exp [W]	PVZP 501 [W]	PVZP C40 [W]	PVL 501 [w]	PVL C40 [w]	PVD [w]	PVZ0 [W]	SUM [W]	Error [%]
200	263.31	114.24	120.50	10.10	5.14	8.31	11.10	269.38	2.31
400	494.70	198.90	209.80	23.21	16.16	16.61	33.12	497.80	0.63
1200	1357.80	478.99	505.24	105.87	81.77	49.83	152.41	1374.13	1.20

Table 26: Experimental coefficient of friction, theoretical coefficient of friction and lubricant factor for PAGD oil.

Speed	Load stage	μ		$\mu_{Michaelis}$		X_L	
		501	C40	501	C40	501	C40
200	K5	0.030	0.033	0.047	0.058	0.648	0.565
	K7	0.025	0.038	0.053	0.066	0.462	0.578
	K9	0.024	0.040	0.058	0.072	0.411	0.555
400	K5	0.018	0.027	0.041	0.050	0.447	0.540
	K7	0.017	0.031	0.046	0.057	0.374	0.536
	K9	0.020	0.032	0.051	0.063	0.386	0.512
1200	K5	0.018	0.032	0.033	0.040	0.537	0.804
	K7	0.011	0.030	0.037	0.046	0.308	0.654
	K9	0.011	0.028	0.041	0.051	0.268	0.546
Average						0.427	0.588

Table 27: PAGD power loss sources at k11 load stage.

Speed [rpm]	PL exp [W]	PVZP 501 [W]	PVZP C40 [W]	PVL 501 [w]	PVL C40 [w]	PVD [w]	PVZ0 [W]	SUM [W]	Error [%]
200	192.44	41.52	87.71	10.95	6.46	8.31	14.69	169.65	11.84
400	345.25	72.30	147.52	27.63	21.16	16.61	39.60	324.82	5.92
1200	1027.14	174.10	336.27	133.26	107.46	49.83	144.09	945.01	8.00

6 Conclusions and Future work

6.1 Conclusions

The results of the tests performed with C40 gear design demonstrated that:

I. The oils that presented lower no-load losses was MINR for lower speed and PAOR for 400 and 1200 rpm;
II. The PAOM presented the highest torque loss for all load stages, when compared with the other PAO's;
III. For low loads(load stage K5): MINE and PAOC have the lowest torque loss for all speed range; PAGD had the highest torque loss for high speeds and MINR for low and medium speeds;
IV. For medium and high loads(load stage K7 and K9), PAGD presented the best performance, despite its high viscosity, and MINR presented the worst performance for all speed range. At these loads, the oils with higher viscosity presented lower torque loss;
V. At high loads and with the increase of the speed the torque loss decrease for all oils;

The results of the test performed with 501 gear design demonstrated that:

I. PAOC when compared with PAO's oils, presented the highest torque loss, so the lowest efficiency, for all load stages;
II. PAOX and ESTR present the highest and the lowest no-load losses, respectively;;
III. At low loads and for all speed range ESTR had the lowest torque loss.
IV. For medium and high loads PAGD distinguishes from the other oils with lower torque loss, despite its higher viscosity;
V. The PAOC oil has in general the worst behaviour among the PAO's for all range speed. At high input load the MINR oil presented the highest torque losses for all speed range;

The results of the tests performed with 951 gear design demonstrated that:

I. The PAOM oil presented lower no-load losses for all speed range, whileat low speeds MINR presented the higher no-load losses and at medium/high speed it was the PAGD oil;
II. PAOR had a good performance at low loads, independently of the input speed, and at medium and high loads it had a good performance at high speeds;
III. For medium and high loads, the MINR oil had the highest torque loss for all speed range. The lowest torque loss for these operating conditions was presented by PAGD oil.

The comparison of the results from the C40 gear design tests with the 501 gear design tests allows to state that::

I. Gear design C40 presented lower no-load losses when lubricated with PAGD, ESTR, PAOC, PAOR and MINE; MINR presented approximately the same behaviour for both 501 and C40 gear designs;
II. PAOC presented lower torque losses with C40 gears for all test conditions;
III. For high load operating conditions the 501 gear geometry, in general, presented lower torque loss for the oils: PAOR, ESTR, PAGD and MINE.
IV. At high speed and low load the geometry C40 has in general lower torque losses, expect for the MINE that had also lower torque loss with the 501 gear geometry.

The comparison of the 501 and 951 gear geometries allowed the fallowing conclusions:

I. For load stage K1 (no-load) the 501 gear design had lower torque losses than the 951 gear design, for all oils tested;
II. At higher loads the 951 gear design always presented lower torque loss than 501 gear design, whatever the oil tested.

6.2 Future work

As future work, it will be interesting to perform the tests with different injection volume flow in order to know the behaviour and the different sources of power loss of each oil type.

Other issue that requires more attention is the sources of power loss. The comprehension of the churning losses in jet lubrication and in bath lubrication, and the seals losses are subjects would deserve more attention.

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Appendix

A.1. Test gearbox Efficiency calculation

The calculation of the FZG efficiency is demonstrated in this chapter. Figure 104 presents a schematic of the FZG test rig

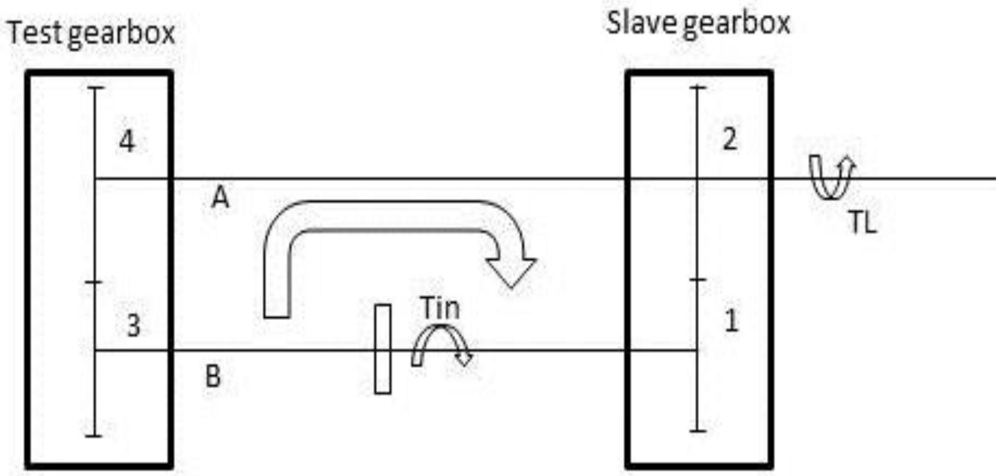

Figure 104: FZG

$$T_A < 1.5T_4 \tag{0.1}$$

$$\eta_{12} = \frac{T_B \times \omega_1}{(T_A + T_L) \times \omega_2} \tag{0.2}$$

$$\eta_{34} = \frac{T_A \times \omega_2}{T_B \times \omega_1} \tag{0.3}$$

$$T_A + T_L = 1.5 \times T_{in} \tag{0.4}$$

$$\eta_{34} = \frac{T_A \times \omega_2}{T_B \times \omega_1} = \frac{(1.5 \times T_{in} - T_L) \times \omega_2}{T_B \times \omega_1} \tag{0.5}$$

$$\frac{\omega_1}{\omega_2} = i \tag{0.6}$$

$$\eta_{12} = \frac{T_B \times \omega_1}{1.5 \times T_{in} \times \omega_2} = \frac{T_B \times \omega_1}{T_{in} \times \omega_1} = \frac{T_B}{T_{in}} \tag{0.7}$$

$$\eta_{34} = \frac{(1.5 \times T_{in} - T_L) \times \omega_2}{\eta_{12} \times T_{in} \times \omega_1} = \frac{1}{\eta_{12}} \times \left(1 - \frac{T_L \times \omega_2}{T_{in} \times \omega_1}\right) \tag{0.8}$$

If slave gearbox equal test gearbox then:

$$\eta_{12} = \eta_{34} = \eta \tag{0.9}$$

Thus:

$$\eta^2 = \left(1 - \frac{T_L \times \omega_2}{T_{in} \times \omega_1}\right) \tag{0.10}$$

$$\eta = \sqrt{\left(1 - \frac{T_L \times \omega_2}{T_{in} \times \omega_1}\right)} \tag{0.11}$$

In the studied case the test gearbox had a 501 design gear and the slave had a C40 design gear.

Calculating previously the C40 efficiency (η_{12}) the value of the 501 efficiency (η_{34}) is given by:

$$\eta_{34} = \frac{1}{\eta_{12}} \times \left(1 - \frac{T_L \times \omega_2}{T_{in} \times \omega_1}\right) \tag{0.12}$$

A.2. Numerical model results

PAOR

- C40

Table 28: Model simulation values of C40 gear design lubricated with PAOR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	11.49	1.28	3.99	8.31	25.07	
	K5	11.49	34.19	10.64	8.31	64.62	0.040
	K7	11.49	71.23	13.21	8.31	104.23	0.044
	K9	11.49	121.36	15.77	8.31	156.92	0.046
400	K1	28.58	2.23	12.12	16.61	59.54	
	K5	28.58	58.53	31.48	16.61	135.21	0.034
	K7	28.58	124.69	38.41	16.61	208.29	0.038
	K9	28.58	209.85	44.94	16.61	299.98	0.040
1200	K1	140.10	5.36	72.13	49.83	267.42	
	K5	140.10	150.51	172.39	49.83	512.83	0.029
	K7	140.10	295.99	208.28	49.83	694.20	0.030
	K9	140.10	490.14	242.41	49.83	922.47	0.031

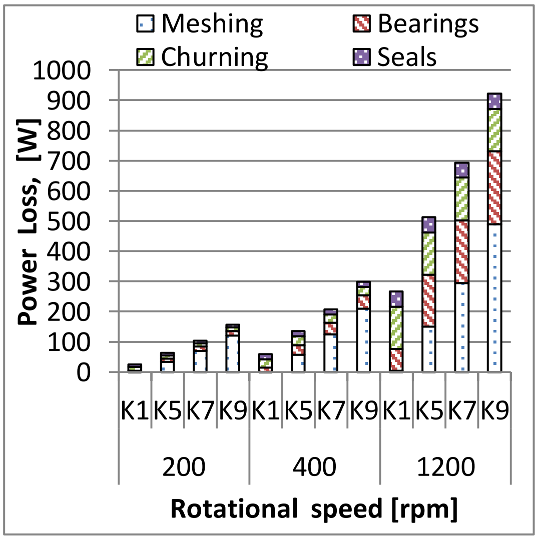

Figure 105: Different sources of C40 gear deign power losses lubricated with PAOR oil.

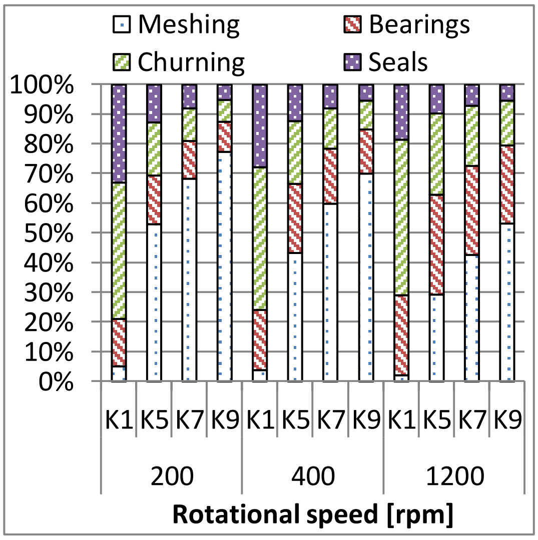

Figure 106: Distribution of the different sources of C40 gear deign power losses lubricated with PAOR oil.

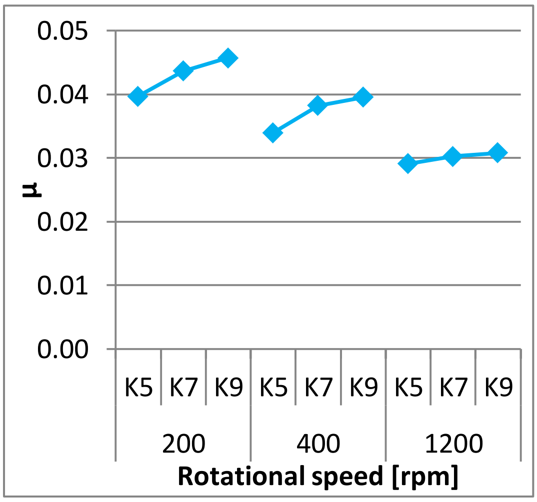

Figure 107: C40 gear design friction coefficient lubricated with PAOR oil.

- 501

Table 29: Model simulation values of 501 gear design lubricated with PAOR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	10.13	0.49	2.33	3.46	16.41	
	K5	10.13	9.58	6.36	3.46	29.54	0.028
	K7	10.13	19.97	9.03	3.46	42.59	0.031
	K9	10.13	47.13	12.99	3.46	73.72	0.045
400	K1	23.09	0.85	5.86	6.92	36.72	
	K5	23.09	15.60	15.25	6.92	60.86	0.023
	K7	23.09	20.92	21.41	6.92	72.34	0.016
	K9	23.09	85.02	29.18	6.92	144.22	0.041
1200	K1	132.05	2.06	29.79	20.76	184.66	
	K5	132.05	81.31	75.34	20.76	213.33	0.058
	K7	132.05	177.15	100.60	20.76	282.91	0.042
	K9	132.05	323.41	129.79	20.76	372.83	0.036

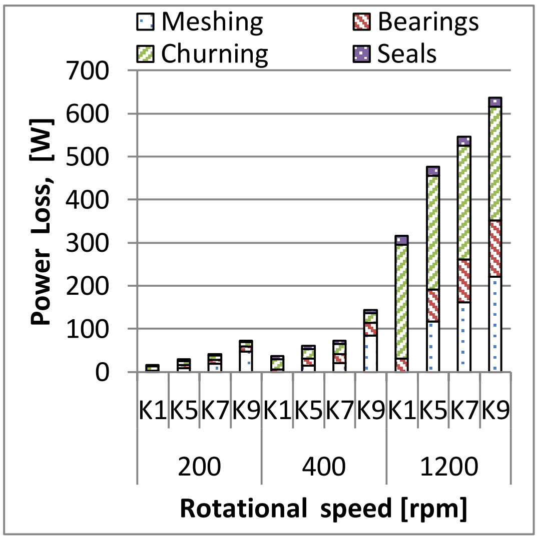

Figure 108: Different sources of 501 gear deign power losses lubricated with PAOR oil.

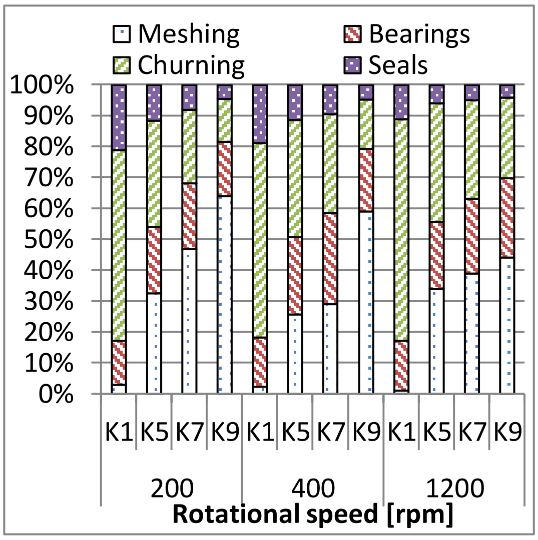

Figure 109: Distribution of the different sources of 501 gear deign power losses lubricated with PAOR oil.

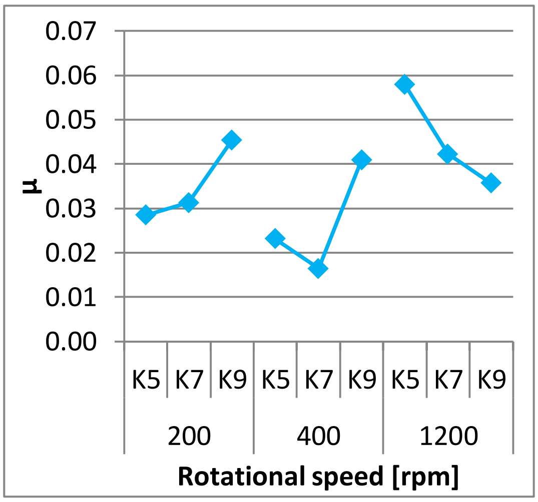

Figure 110: 501 gear design friction coefficient lubricated with PAOR oil.

- 951

Table 30: Model simulation values of 951 gear design lubricated with PAOR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	Auxiliary loss [W]	μ
200	K1	6.95	0.26	2.40	3.46	13.07	6.95	
	K5	6.95	10.47	6.96	3.46	27.83		0.054
	K7	6.95	22.82	10.52	3.46	43.75		0.062
	K9	6.95	41.65	15.44	3.46	67.50		0.069
400	K1	17.55	0.46	5.88	6.92	30.80	17.55	
	K5	17.55	18.22	16.93	6.92	59.62		0.047
	K7	17.55	39.70	24.31	6.92	88.48		0.054
	K9	17.55	72.47	33.86	6.92	130.79		0.060
1200	K1	144.22	1.11	31.64	20.76	197.73	144.22	
	K5	144.22	43.87	82.59	20.76	291.44		0.037
	K7	144.22	95.66	111.06	20.76	371.70		0.043
	K9	144.22	174.59	145.96	20.76	485.52		0.048

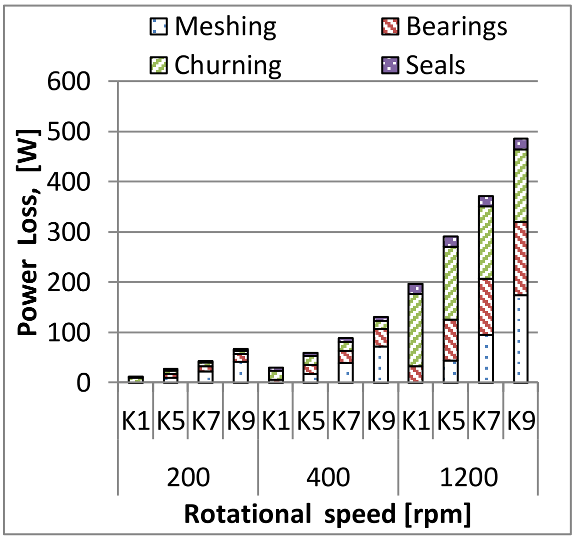

Figure 111: Different sources of 951 gear deign power losses lubricated with PAOR oil.

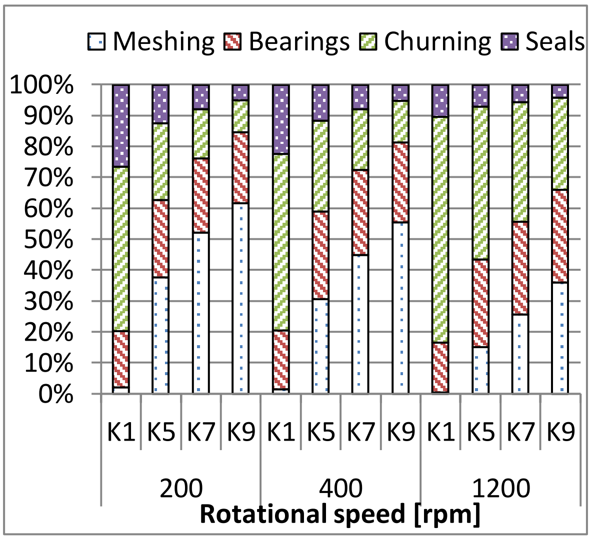

Figure 112: Distribution of the Different sources of 951 gear deign power losses lubricated with PAOR oil.

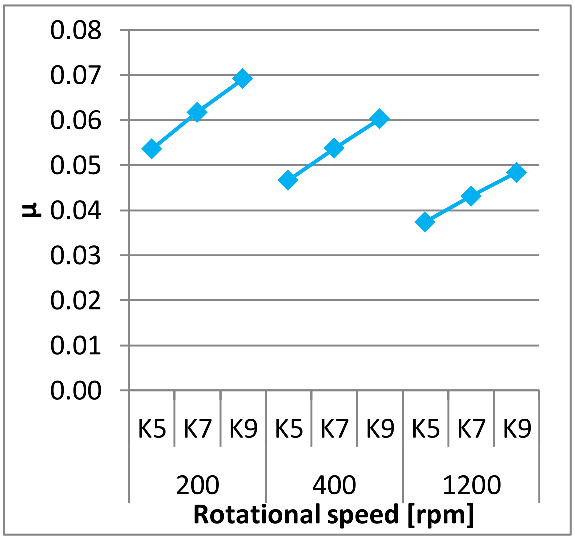

Figure 113: 951 gear design friction coefficient lubricated with PAOR oil.

PAOX

- C40

Table 31: Model simulation values of C40 gear design lubricated with PAOX oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	16.05	1.27	4.34	8.31	29.97	0.031
	K5	16.05	31.35	11.15	8.31	66.86	0.036
	K7	16.05	69.90	14.65	8.31	108.91	0.043
	K9	16.05	114.59	16.37	8.31	155.32	0.043
400	K1	44.25	2.22	12.61	16.61	75.70	0.027
	K5	44.25	49.13	31.80	16.61	141.79	0.029
	K7	44.25	110.41	39.03	16.61	210.30	0.034
	K9	44.25	201.98	45.76	16.61	308.61	0.038
1200	K1	182.76	5.36	72.85	49.83	310.80	0.022
	K5	182.76	95.60	175.02	49.83	503.21	0.018
	K7	182.76	241.73	209.41	49.83	683.74	0.025
	K9	182.76	455.05	242.37	49.83	930.01	0.029

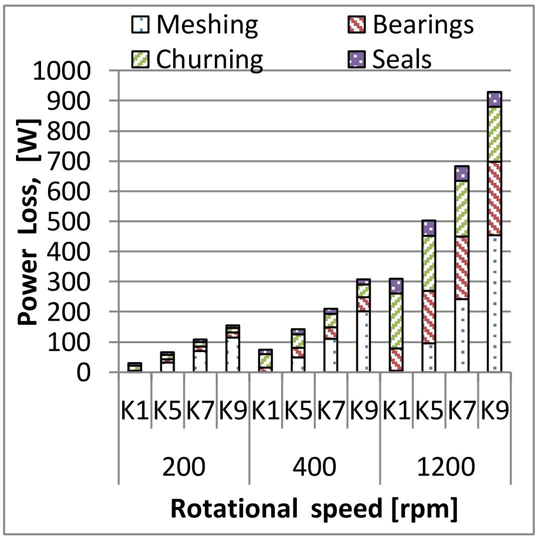

Figure 114: Different sources of 501 gear deign power losses lubricated with PAOX oil.

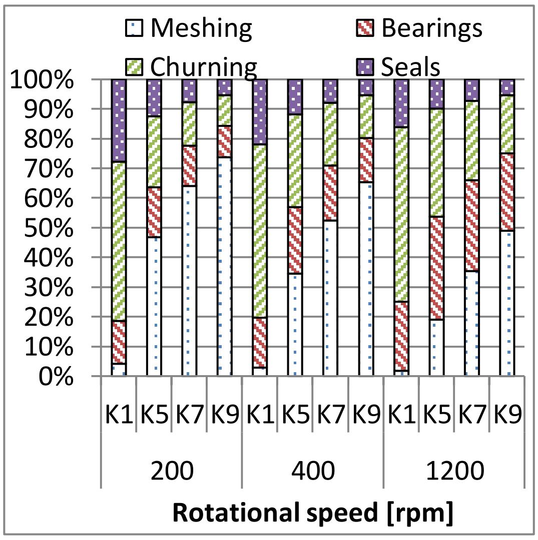

Figure 115: Distribution of the different sources of C40 gear deign power losses lubricated with PAOX oil.

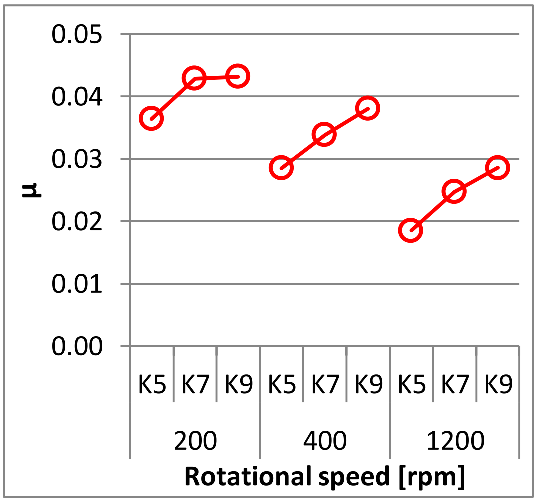

Figure 116: C40 gear design friction coefficient lubricated with PAOX oil.

- 501

Table 32: Model simulation values of 501 gear design lubricated with PAOX oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	1.85	0.49	2.31	3.46	8.11	0.031
	K5	1.85	11.66	6.32	3.46	23.30	0.035
	K7	1.85	33.07	9.13	3.46	47.52	0.052
	K9	1.85	60.49	12.99	3.46	78.79	0.058
400	K1	7.00	0.85	5.64	6.92	20.41	0.027
	K5	7.00	21.36	15.49	6.92	50.77	0.032
	K7	7.00	77.01	21.45	6.92	112.38	0.060
	K9	7.00	107.27	29.27	6.92	150.47	0.052
1200	K1	28.38	2.05	30.61	20.76	81.80	0.021
	K5	28.38	84.39	75.79	20.76	209.33	0.0417
	K7	28.38	174.57	101.81	20.76	325.52	0.046
	K9	28.38	252.38	130.65	20.76	432.18	0.041

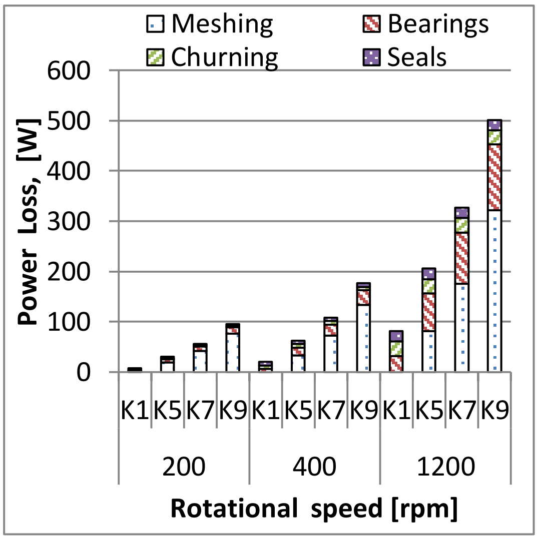

Figure 117: Different sources of 501 gear deign power losses lubricated with PAOX oil.

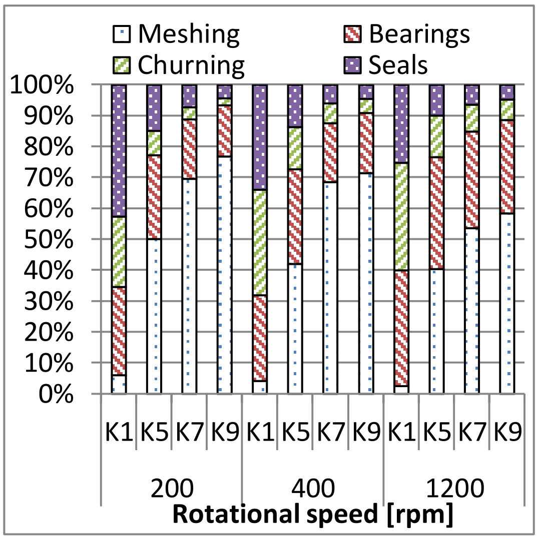

Figure 118: Distribution of the different sources of 501 gear deign power losses lubricated with PAOX oil.

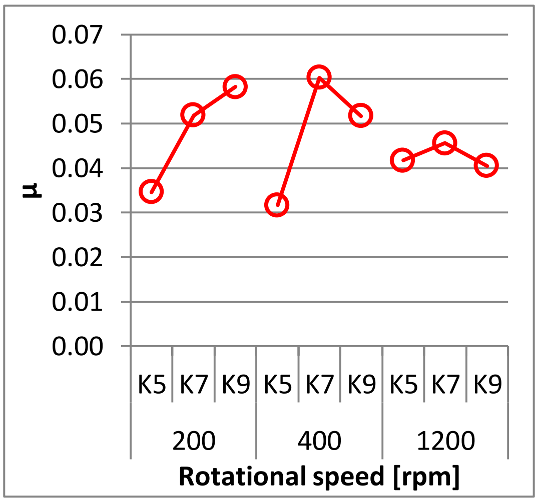

Figure 119: 501 gear design friction coefficient lubricated with PAOX oil.

PAOC

- C40

Table 33: Model simulation values of C40 gear design lubricated with PAOC oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	µ
200	K1	12.75	1.27	4.34	8.31	26.67	
	K5	12.75	30.12	11.15	8.31	62.33	0.035
	K7	12.75	67.84	14.65	8.31	103.55	0.042
	K9	12.75	125.48	16.37	8.31	162.91	0.047
400	K1	31.50	2.22	12.61	16.61	62.94	
	K5	31.50	50.84	31.80	16.61	130.75	0.030
	K7	31.50	123.42	39.03	16.61	210.56	0.038
	K9	31.50	219.07	45.76	16.61	312.95	0.041
1200	K1	153.18	5.36	72.85	49.83	281.22	
	K5	153.18	115.45	175.02	49.83	493.48	0.022
	K7	153.18	284.23	209.41	49.83	696.65	0.029
	K9	153.18	497.65	242.37	49.83	943.03	0.031

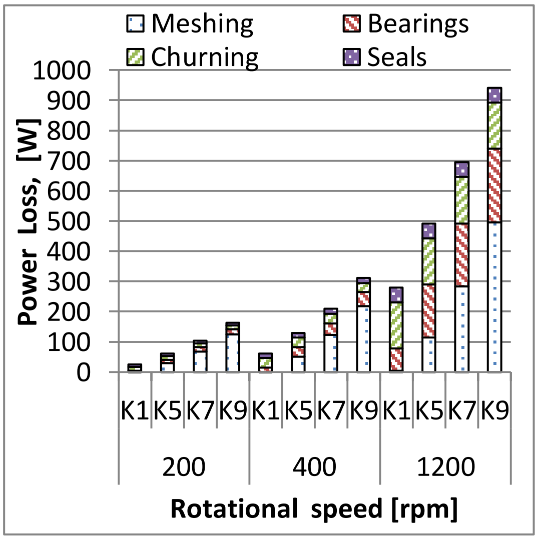

Figure 120: Different sources of C40 gear deign power losses lubricated with PAOC oil.

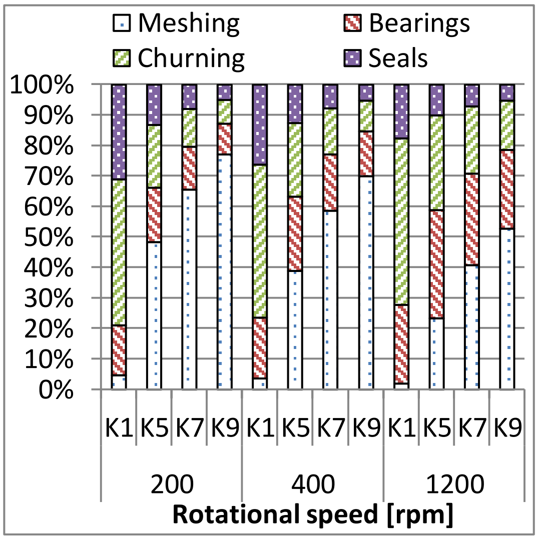

Figure 121: Distribution of the different sources of C40 gear deign power losses lubricated with PAOC oil.

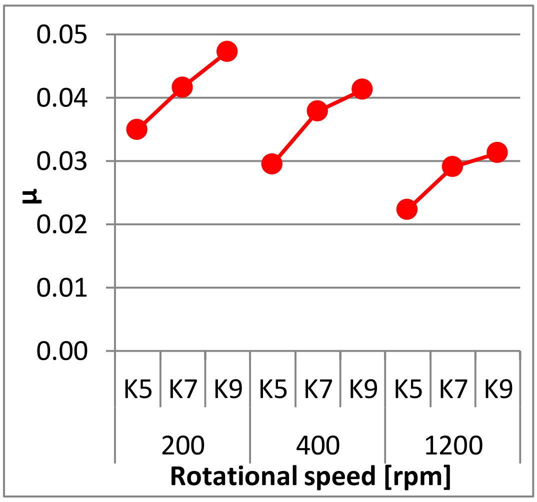

Figure 122: C40 gear design friction coefficient lubricated with PAOC oil.

- 501

Table 34: Model simulation values of 501 gear design lubricated with PAOC oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	7.59	0.49	2.30	3.46	13.84	
	K5	7.59	29.06	6.24	3.46	46.36	0.086
	K7	7.59	49.40	9.13	3.46	69.59	0.077
	K9	7.59	61.68	12.96	3.46	85.69	0.059
400	K1	37.82	0.84	5.96	6.92	32.64	
	K5	37.82	31.48	15.50	6.92	72.81	0.047
	K7	37.82	82.47	21.83	6.92	130.12	0.065
	K9	37.82	86.07	29.06	6.92	140.97	0.041
1200	K1	276.10	2.06	29.53	20.76	190.41	
	K5	276.10	134.81	74.67	20.76	368.30	0.067
	K7	276.10	185.63	98.93	20.76	443.38	0.048
	K9	276.10	237.18	128.54	20.76	524.53	0.038

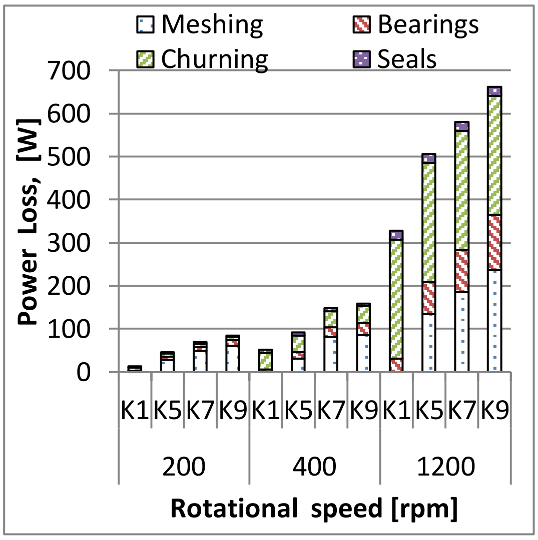

Figure 123: Different sources of 501 gear deign power losses lubricated with PAOC oil.

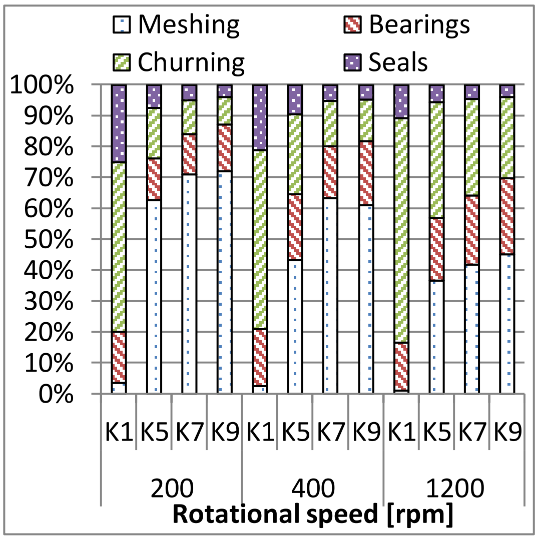

Figure 124: Distribution of the Different sources of 501 gear deign power losses lubricated with PAOC oil.

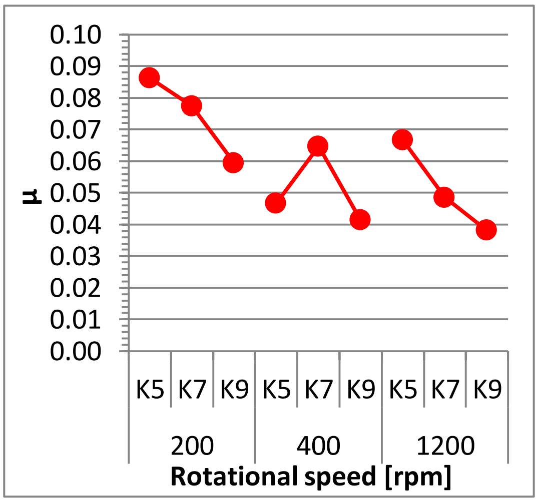

Figure 125: 501 gear design friction coefficient lubricated with PAOC oil.

PAOM

- C40

Table 35: Model simulation values of C40 gear design lubricated with PAOM oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	20.18	1.28	4.20	8.31	33.96	
	K5	20.18	31.01	11.15	8.301	70.64	0.036
	K7	20.18	70.87	13.80	8.31	113.15	0.043
	K9	20.18	125.99	16.28	8.31	170.76	0.047
400	K1	47.16	2.22	12.62	16.61	78.61	
	K5	47.16	51.45	32.64	16.61	147.86	0.030
	K7	47.16	120.83	40.02	16.61	224.62	0.037
	K9	47.16	213.38	46.41	16.61	323.56	0.040
1200	K1	165.52	5.35	74.17	49.83	294.86	
	K5	165.52	136.98	177.21	49.83	529.53	0.027
	K7	165.52	298.76	213.87	49.83	727.98	0.031
	K9	165.52	504.53	247.15	49.83	967.03	0.032

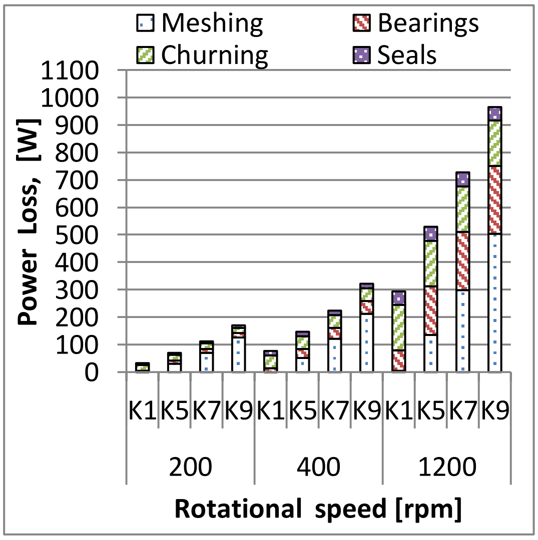

Figure 126: Different sources of C40 gear deign power losses lubricated with PAOM oil.

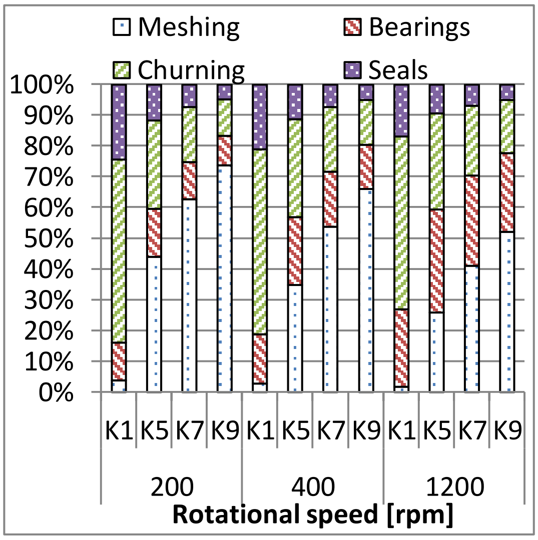

Figure 127: Distribution of the different sources of C40 gear deign power losses lubricated with PAOM oil.

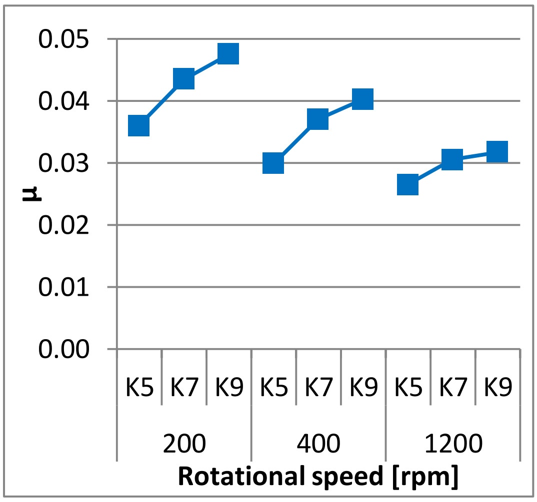

Figure 128: C40 gear design friction coefficient lubricated with PAOM oil.

- 501

Table 36: Model simulation values of 501 gear design lubricated with PAOM oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	7.55	0.49	2.35	3.46	13.84	
	K5	7.55	21.61	6.29	3.46	38.91	0.064
	K7	7.55	33.06	9.17	3.46	53.25	0.052
	K9	7.55	30.96	13.06	3.46	55.03	0.030
400	K1	17.28	0.85	5.64	6.92	30.68	
	K5	17.28	36.85	15.69	6.92	76.74	0.055
	K7	17.28	53.75	21.79	6.92	99.74	0.042
	K9	17.28	55.27	29.67	6.92	109.14	0.027
1200	K1	240.88	2.05	30.60	20.76	173.86	
	K5	240.88	18.09	78.00	20.76	237.29	0.009
	K7	240.88	88.90	102.56	20.76	332.67	0.023
	K9	240.88	61.45	132.23	20.76	334.88	0.010

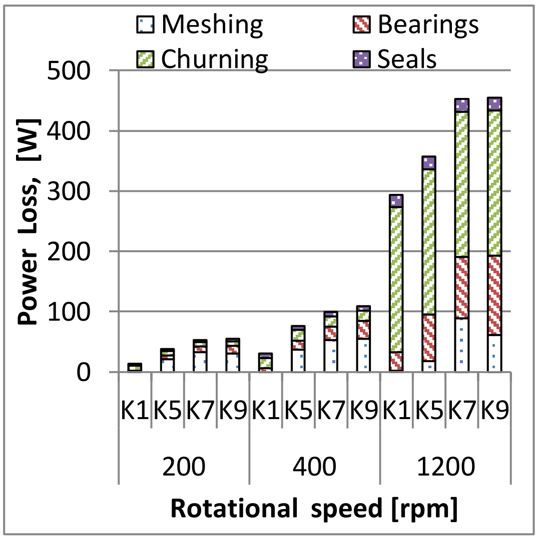

Figure 129: Different sources of 501 gear deign power losses lubricated with PAOM oil.

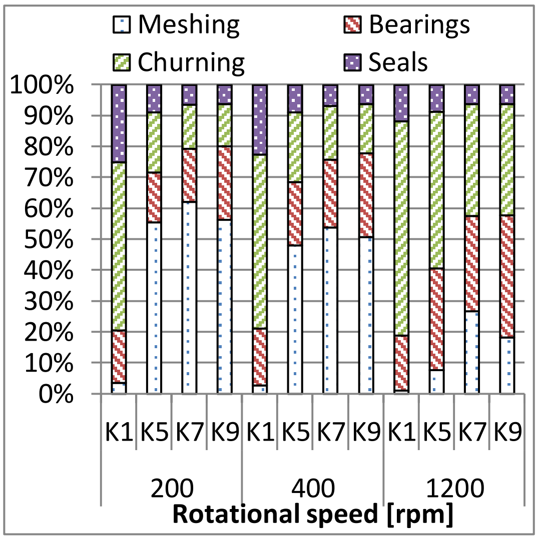

Figure 130: Distribution of the Different sources of 501 gear deign power losses lubricated with PAOM oil.

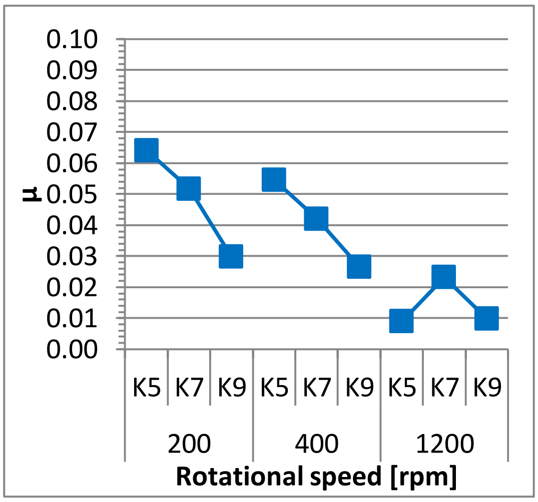

Figure 131: 501 gear design friction coefficient lubricated with PAOM oil.

MINE

- C40

Table 37: Model simulation values of c40 gear design lubricated with MINE oil

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	12.21	1.30	4.26	8.31	26.07	
	K5	12.21	30.38	11.30	8.31	62.20	0.035
	K7	12.21	76.74	14.09	8.31	111.34	0.047
	K9	12.21	131.98	16.67	8.31	169.16	0.050
400	K1	33.51	2.26	12.81	16.61	65.19	
	K5	33.51	48.34	32.98	16.61	131.44	0.028
	K7	33.51	130.26	40.39	16.61	220.77	0.040
	K9	33.51	234.30	47.63	16.61	332.05	0.044
1200	K1	148.79	5.45	74.80	49.83	278.87	
	K5	148.79	108.14	180.21	49.83	486.97	0.021
	K7	148.79	256.77	216.99	49.83	672.39	0.026
	K9	148.79	534.98	251.98	49.83	985.58	0.034

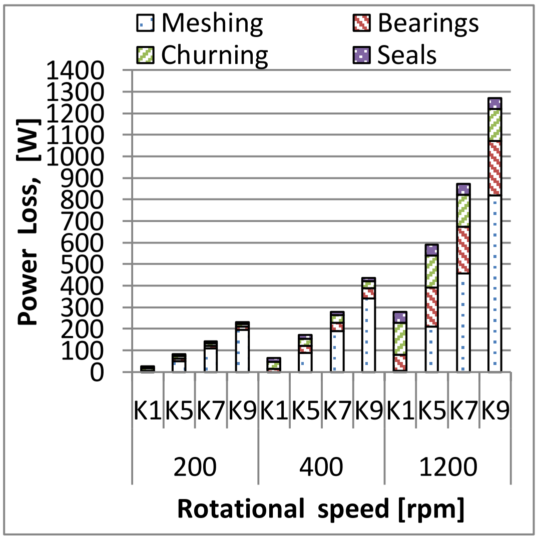

Figure 132: Different sources of C40 gear deign power losses lubricated with MINE oil.

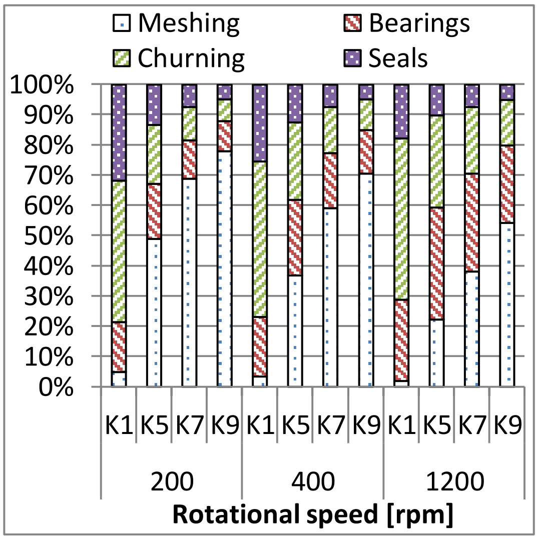

Figure 133: Distribution of the different sources of C40 gear deign power losses lubricated with MINE oil.

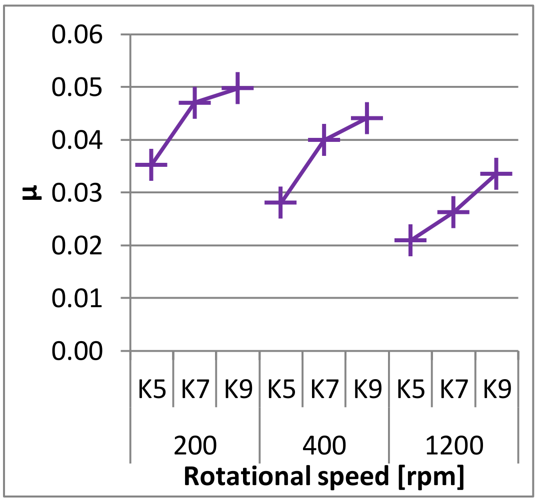

Figure 134: C40 gear design friction coefficient lubricated with MINE oil.

- 501

Table 38: Model simulation values of 501 gear design lubricated with MINE oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	Auxiliary loss [W]	μ
200	K1	12.02	0.49	2.37	3.46	18.34	0.00	
	K5	12.02	6.46	6.45	3.46	28.40		0.019
	K7	12.02	33.96	9.43	3.46	58.88		0.053
	K9	12.02	49.12	13.47	3.46	78.07		0.047
400	K1	13.20	0.85	5.73	6.92	26.70	13.20	
	K5	13.20	12.74	15.90	6.92	48.76		0.019
	K7	13.20	63.38	22.17	6.92	105.66		0.050
	K9	13.20	93.48	30.03	6.92	143.63		0.045
1200	K1	112.80	2.04	30.85	20.76	166.45	0.00	
	K5	112.80	26.52	78.44	20.76	238.51		0.013
	K7	112.80	138.71	104.11	20.76	376.39		0.036
	K9	112.80	163.98	134.29	20.76	431.83		0.026

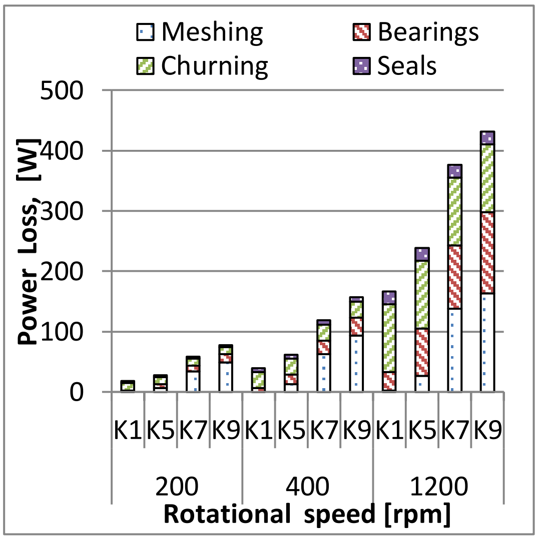

Figure 135: Different sources of 501 gear deign power losses lubricated with MINE oil.

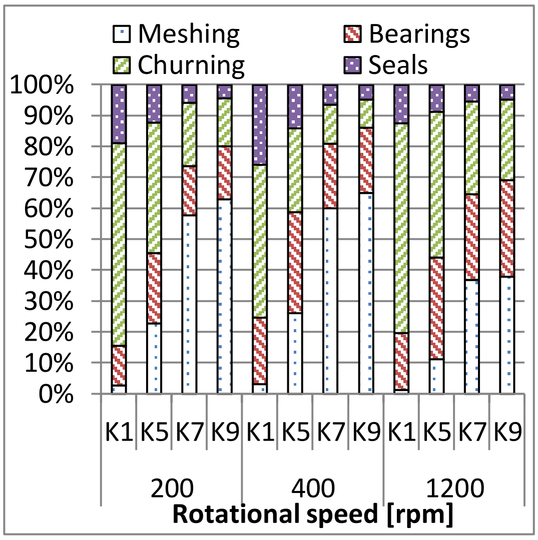

Figure 136: Distribution of the different sources of 501 gear deign power losses lubricated with MINE oil.

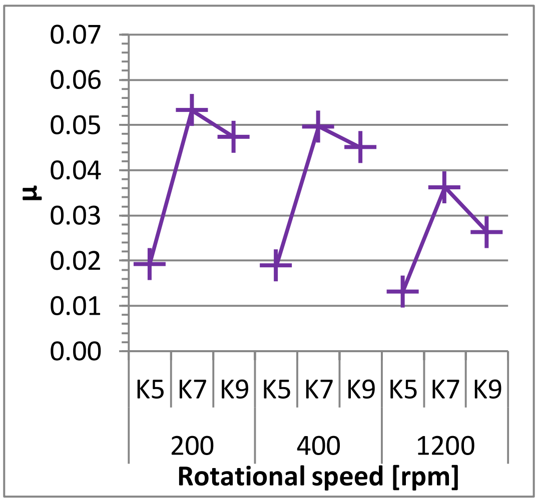

Figure 137: 501 gear design friction coefficient lubricated with MINE oil.

MINR

- C40

Table 39: Model simulation values of C40 gear design lubricated with MINR oil

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	11.10	1.33	3.33	8.31	24.07	
	K5	11.10	48.97	8.96	8.31	77.34	0.057
	K7	11.10	92.58	11.27	8.31	123.25	0.057
	K9	11.10	152.91	13.60	8.31	185.92	0.058
400	K1	33.12	2.31	10.23	16.61	62.27	
	K5	33.12	81.43	26.22	16.61	157.38	0.047
	K7	33.12	157.72	32.77	16.61	240.22	0.048
	K9	33.12	268.75	38.63	16.61	357.11	0.051
1200	K1	152.41	5.57	61.52	49.83	269.34	
	K5	152.41	178.69	148.92	49.83	529.85	0.035
	K7	152.41	346.39	180.08	49.83	728.72	0.035
	K9	152.41	603.78	208.95	49.83	1014.98	0.038

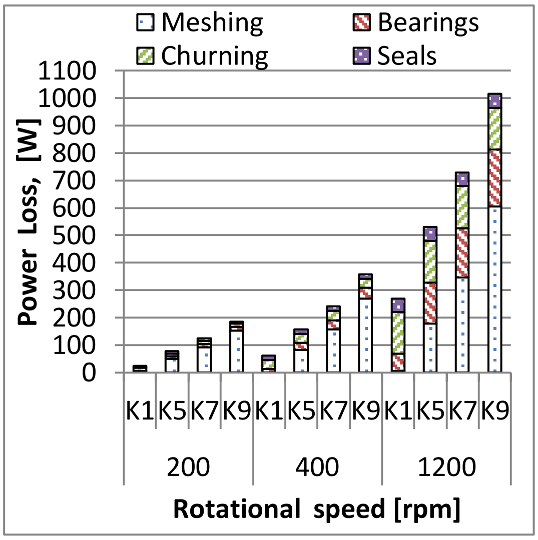

Figure 138: Different sources of C40 gear deign power losses lubricated with MINR oil.

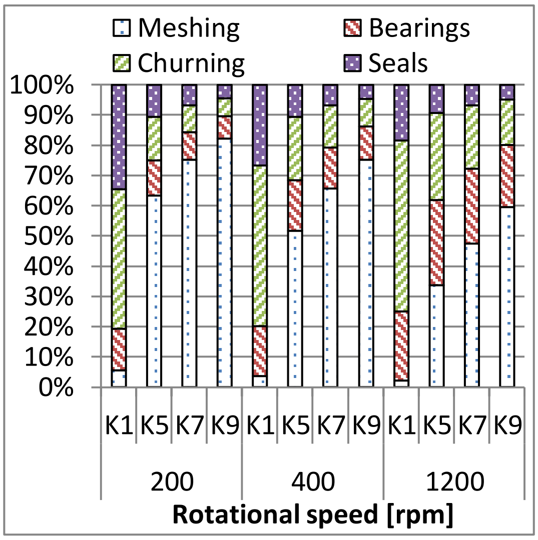

Figure 139: Distribution of the different sources of C40 gear deign power losses lubricated with MINR oil.

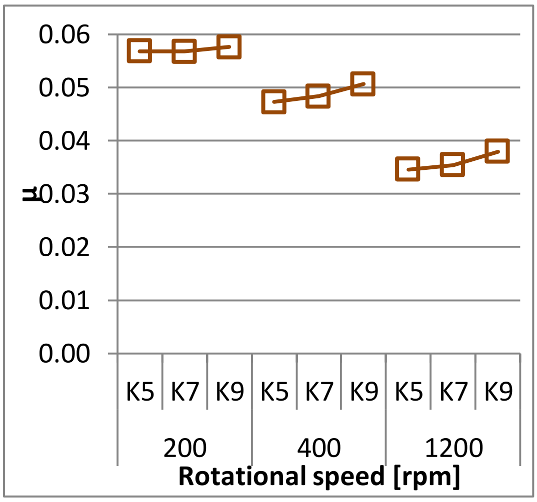

Figure 140: C40 gear design friction coefficient lubricated with MINR oil.

- 501

Table 40: Model simulation values of 501 gear design lubricated with MINR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	6.23	0.50	2.26	3.46	12.45	
	K5	6.23	15.60	5.99	3.46	31.29	0.046
	K7	6.23	38.91	9.05	3.46	57.66	0.061
	K9	6.23	70.44	13.30	3.46	93.43	0.068
400	K1	16.70	0.87	5.12	6.92	29.60	
	K5	16.70	30.79	13.87	6.92	68.28	0.046
	K7	16.70	69.44	19.89	6.92	112.94	0.054
	K9	16.70	125.94	27.74	6.92	177.30	0.061
1200	K1	73.61	2.09	26.95	20.76	123.41	
	K5	73.61	85.04	66.61	20.76	246.02	0.042
	K7	73.61	178.27	88.92	20.76	361.57	0.047
	K9	73.61	313.58	115.98	20.76	523.93	0.050

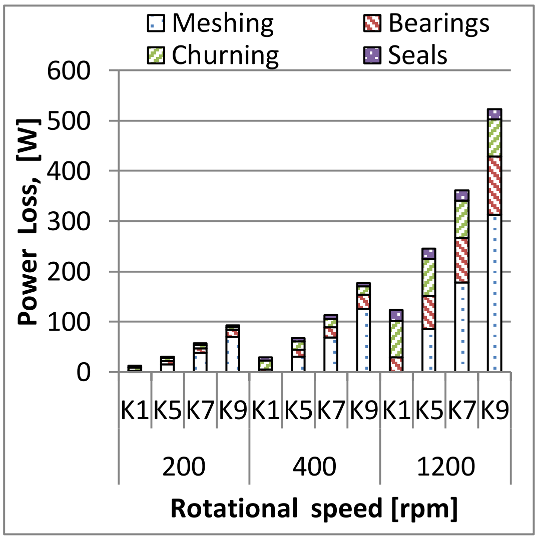

Figure 141: Different sources of 501 gear deign power losses lubricated with MINR oil.

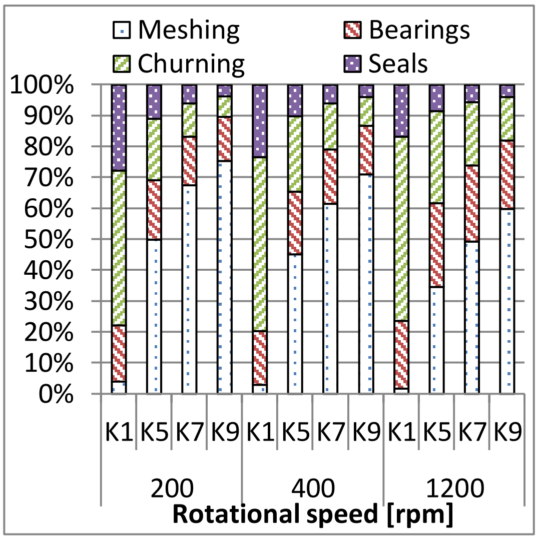

Figure 142: Distribution of the different sources of 501 gear deign power losses lubricated with MINR oil.

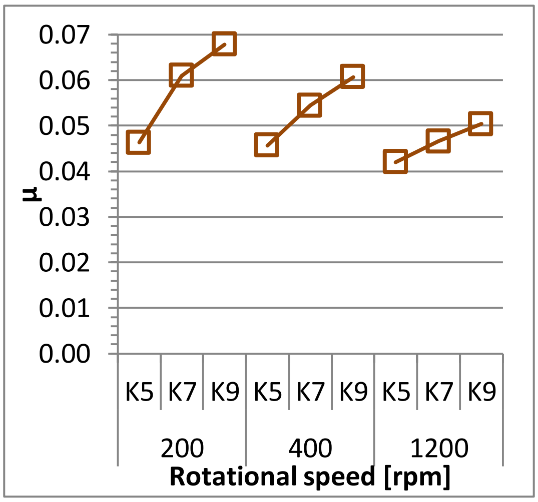

Figure 143: 501 gear design friction coefficient lubricated with MINR oil.

- 951

Table 41: Model simulation values of 951 gear design lubricated with MINR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	Auxiliary loss [W]	μ
200	K1	6.23	0.27	2.38	3.46	12.34	14.60	
	K5	6.23	10.63	6.86	3.46	27.18		0.054
	K7	6.23	23.25	10.63	3.46	43.57		0.063
	K9	6.23	42.54	16.04	3.46	68.27		0.071
400	K1	16.70	0.47	5.42	6.92	29.51	27.02	
	K5	16.70	18.56	15.52	6.92	57.69		0.047
	K7	16.70	40.50	22.83	6.92	86.94		0.055
	K9	16.70	74.07	32.54	6.92	130.22		0.062
1200	K1	73.61	1.13	28.32	20.76	123.82	104.40	
	K5	73.61	44.69	72.99	20.76	212.05		0.038
	K7	73.61	97.54	99.21	20.76	291.13		0.044
	K9	73.61	178.35	131.23	20.76	403.95		0.049

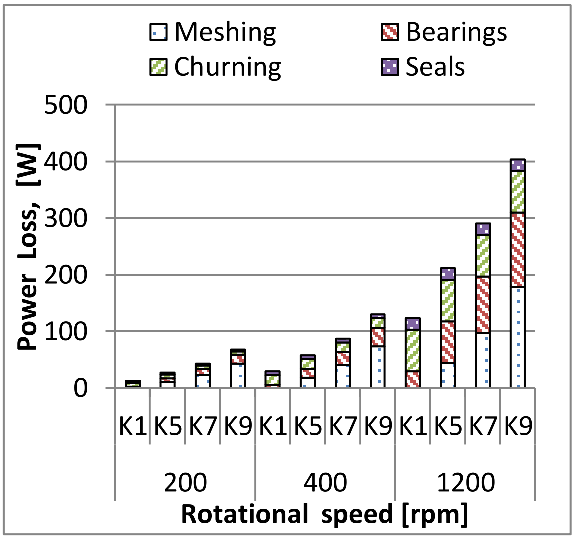

Figure 144: Different sources of 951 gear deign power losses lubricated with MINR oil.

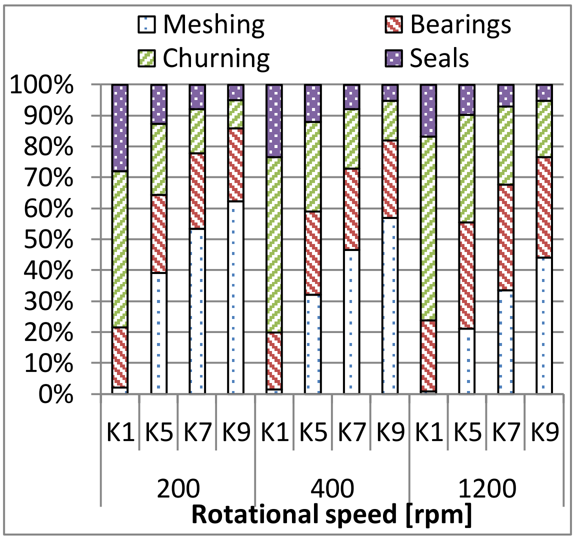

Figure 145: Distribution of the Different sources of 951 gear deign power losses lubricated with MINR oil.

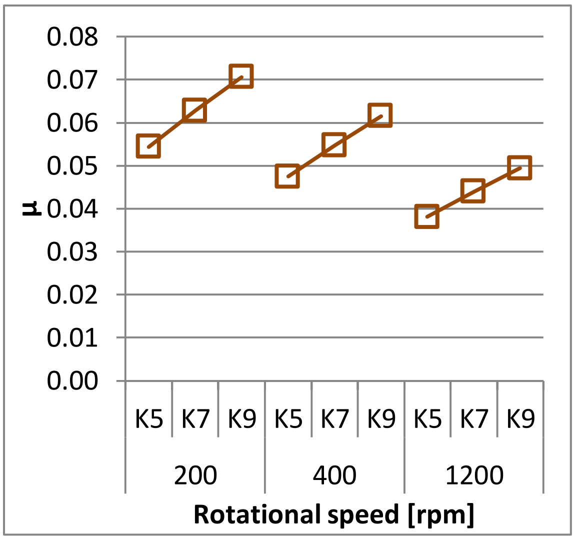

Figure 146: 951 gear design friction coefficient lubricated with MINR oil.

PAGD

- C40

Table 42: Model simulation values of C40 gear design lubricated with PAGD oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	14.69	1.25	4.64	8.31	28.89	
	K5	14.69	28.14	12.20	8.31	63.33	0.033
	K7	14.69	61.85	15.03	8.31	99.88	0.038
	K9	14.69	106.60	17.58	8.31	147.18	0.040
400	K1	39.60	2.18	14.10	16.61	72.48	
	K5	39.60	46.82	36.03	16.61	139.06	0.027
	K7	39.60	99.92	44.21	16.61	200.34	0.031
	K9	39.60	171.13	51.66	16.61	278.99	0.032
1200	K1	144.09	5.24	80.44	49.83	279.60	
	K5	144.09	167.73	193.82	49.83	555.48	0.032
	K7	144.09	293.74	234.77	49.83	722.42	0.030
	K9	144.09	439.69	271.75	49.83	905.36	0.028

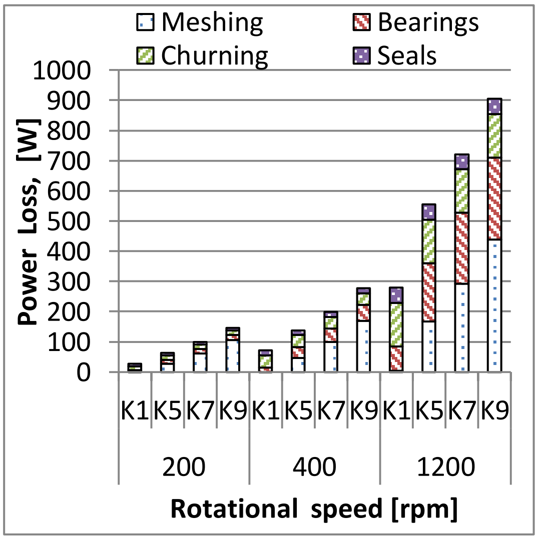

Figure 147: Different sources of C40 gear deign power losses lubricated with PAGD oil.

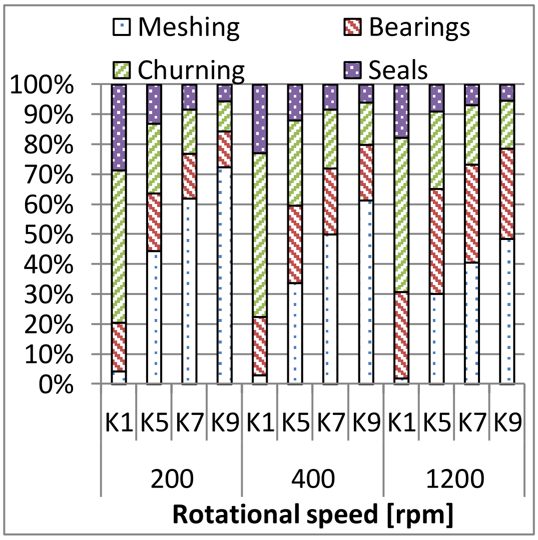

Figure 148: Distribution of the different sources of C40 gear deign power losses lubricated with PAGD oil.

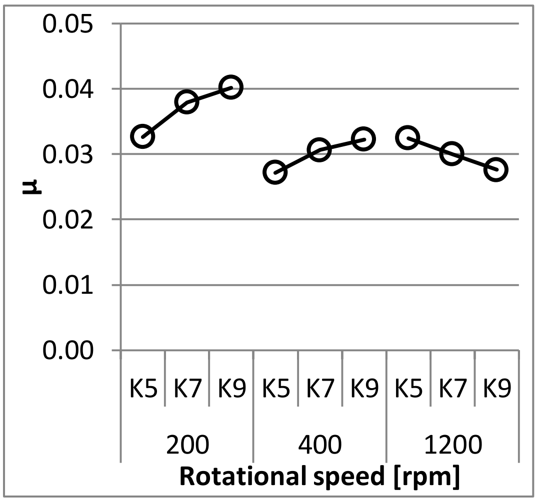

Figure 149: C40 gear design friction coefficient lubricated with PAGD oil.

- 501

Table 43: Model simulation values of 501 gear design lubricated with PAGD oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	10.12	0.47	2.43	3.46	16.49	
	K5	10.12	10.20	6.74	3.46	30.53	0.030
	K7	10.12	15.65	9.36	3.46	38.59	0.025
	K9	10.12	24.95	13.09	3.46	51.63	0.024
400	K1	23.57	0.83	6.12	6.92	37.44	
	K5	23.57	12.26	16.93	6.92	59.68	0.018
	K7	23.57	22.06	23.36	6.92	75.91	0.017
	K9	23.57	40.80	31.39	6.92	102.68	0.020
1200	K1	108.89	2.00	32.72	20.76	164.36	
	K5	108.89	35.44	83.77	20.76	248.86	0.0175
	K7	108.89	43.74	110.67	20.76	284.06	0.011
	K9	108.89	68.24	141.98	20.76	339.87	0.011

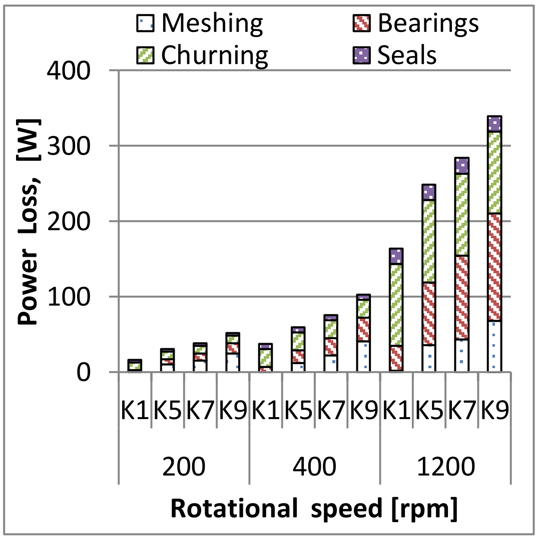

Figure 150: Different sources of 501 gear deign power losses lubricated with PAGD oil.

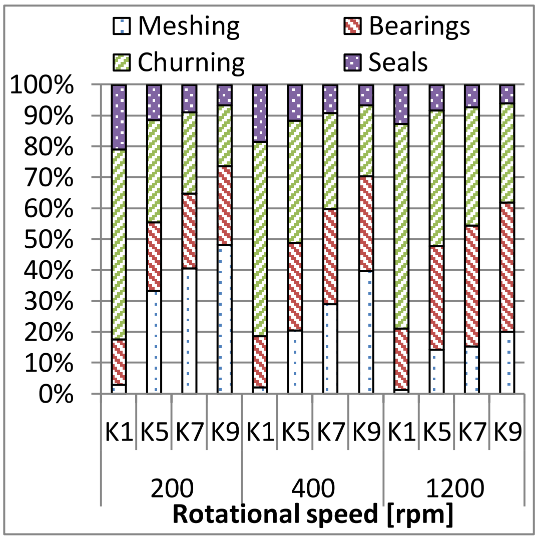

Figure 151: Distribution of the different sources of 501 gear deign power losses lubricated with PAGD oil.

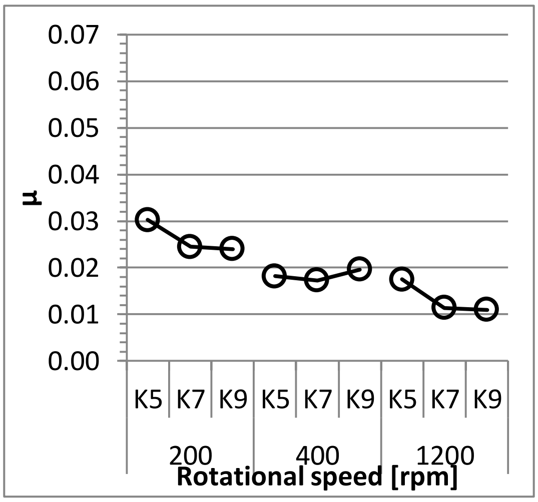

Figure 152: 501 gear design friction coefficient lubricated with PAGD oil.

- 951

Table 44: Model simulation values of 501 gear design lubricated with PAGD oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	Auxiliary loss [W]	µ
200	K1	10.12	0.26	2.50	3.46	16.34	6.31	
	K5	10.12	10.15	7.31	3.46	31.05		0.052
	K7	10.12	22.08	10.79	3.46	46.46		0.060
	K9	10.12	40.24	15.41	3.46	69.24		0.067
400	K1	23.57	0.45	6.44	6.92	37.38	22.29	
	K5	23.57	17.67	18.79	6.92	66.95		0.045
	K7	23.57	38.47	26.36	6.92	95.32		0.052
	K9	23.57	70.09	36.00	6.92	136.58		0.058
1200	K1	108.89	1.08	34.46	20.76	165.18	303.75	
	K5	108.89	42.57	91.48	20.76	263.70		0.036
	K7	108.89	92.64	122.66	20.76	344.95		0.042
	K9	108.89	168.77	159.57	20.76	457.99		0.047

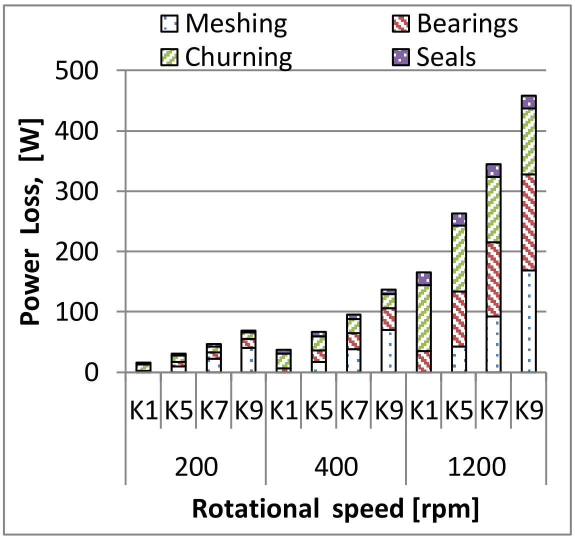

Figure 153: Different sources of 951 gear deign power losses lubricated with PAGD oil.

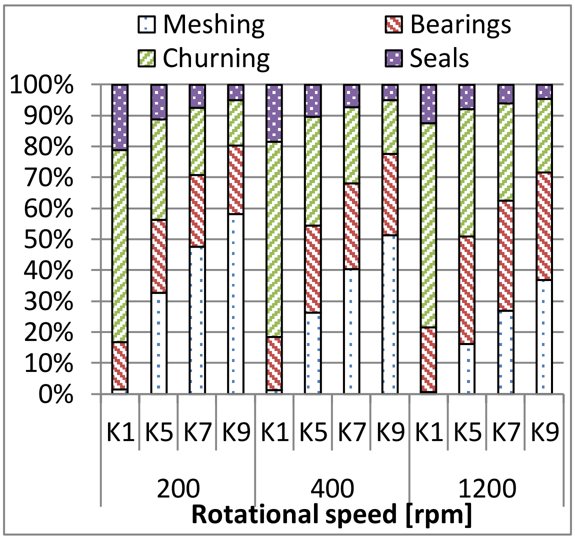

Figure 154: Distribution of the different sources of 951 gear deign power losses lubricated with PAGD oil.

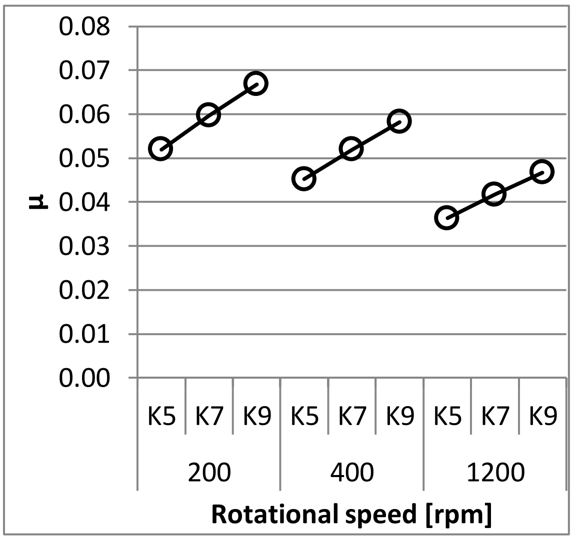

Figure 155: 951 gear design friction coefficient lubricated with PAGD oil.

ESTR

- C40

Table 45: Model simulation values of C40 gear design lubricated with ESTR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]	μ
200	K1	12.77	1.28	3.86	8.31	26.22	
	K5	12.77	34.33	10.28	8.31	65.69	0.040
	K7	12.77	68.16	12.73	8.31	101.97	0.042
	K9	12.77	124.37	15.10	8.31	160.54	0.047
400	K1	33.43	2.23	11.74	16.61	64.01	
	K5	33.43	58.81	30.11	16.61	138.96	0.034
	K7	33.43	116.18	36.96	16.61	203.17	0.036
	K9	33.43	213.49	43.33	16.61	306.85	0.040
1200	K1	151.21	5.37	69.72	49.83	276.14	
	K5	151.21	148.64	166.68	49.83	516.36	0.029
	K7	151.21	270.53	199.94	49.83	671.52	0.028
	K9	151.21	486.13	231.53	49.83	918.72	0.031

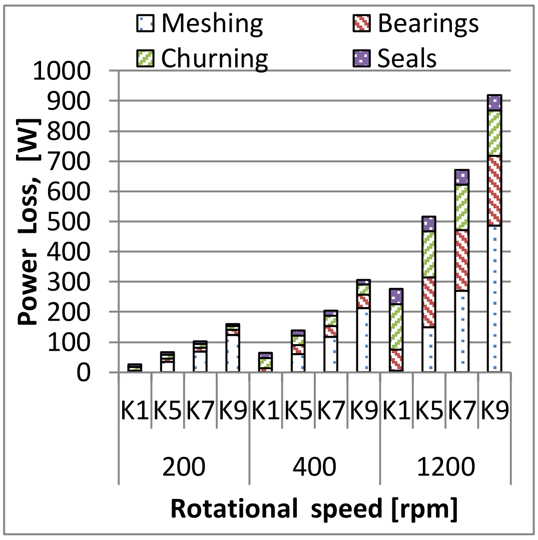

Figure 156: Different sources of C40 gear deign power losses lubricated with ESTR oil.

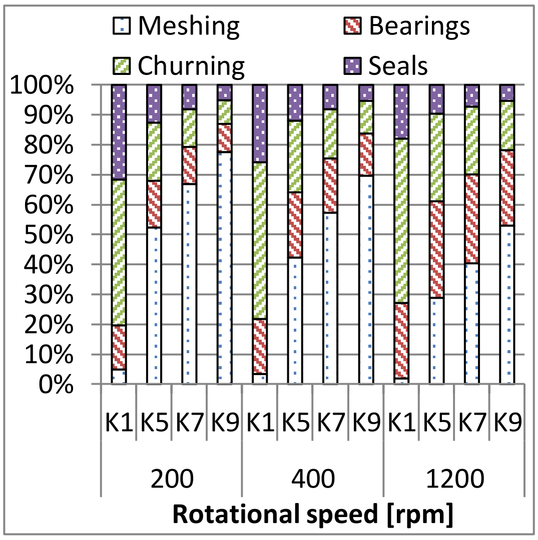

Figure 157: Distribution of the Different sources of C40 gear deign power losses lubricated with ESTR oil.

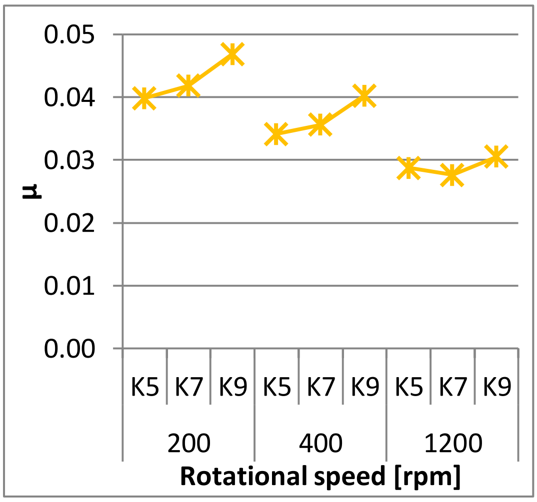

Figure 158: C40 gear design friction coefficient lubricated with ESTR oil.

- 501

Table 46: Model simulation values of 501 lubricated with ESTR oil.

Speed	Load stage	Churning [W]	Meshing [W]	Bearings [W]	Seals [W]	Power Loss [W]		µ
200	K1	9.09	0.49	2.37	3.46	15.42	9.09	0.031
	K5	9.09	2.83	6.40	3.46	21.78		0.008
	K7	9.09	21.34	9.52	3.46	43.42		0.033
	K9	9.09	65.92	13.83	3.46	92.30		0.063
400	K1	19.90	0.85	5.48	6.92	33.15	19.90	0.027
	K5	19.90	5.29	15.09	6.92	47.21		0.008
	K7	19.90	36.29	21.24	6.92	84.35		0.028
	K9	19.90	110.40	29.08	6.92	166.30		0.053
1200	K1	74.07	2.06	29.17	20.76	126.05	74.07	0.022
	K5	74.07	25.48	73.48	20.76	193.78		0.013
	K7	74.07	96.24	97.71	20.76	288.78		0.025
	K9	74.07	261.98	125.94	20.76	482.76		0.042

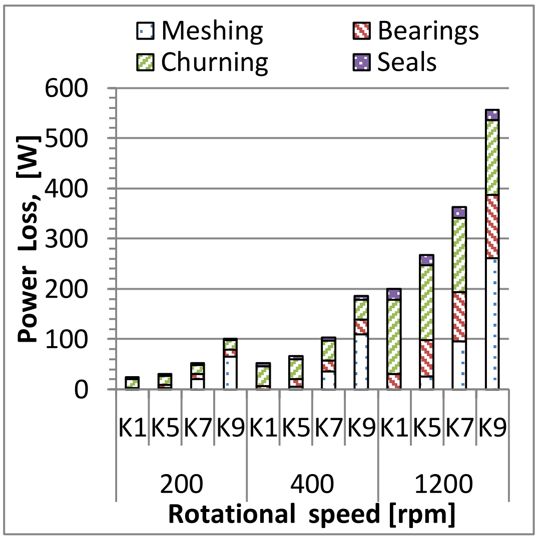

Figure 159: Different sources of 501 gear deign power losses lubricated with ESTR oil.

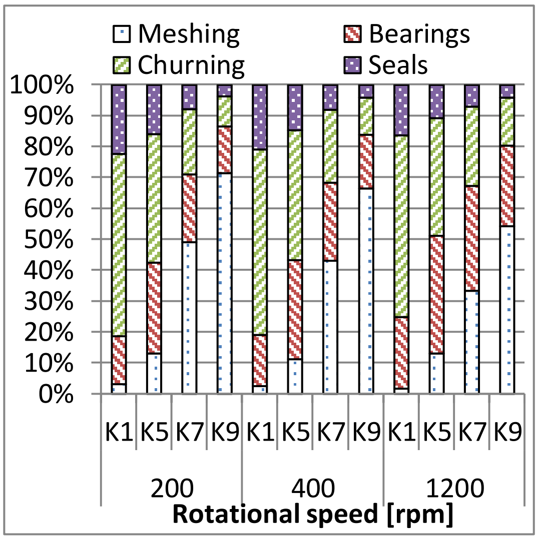

Figure 160: Distribution of the different sources of 501 gear deign power losses lubricated with ESTR oil.

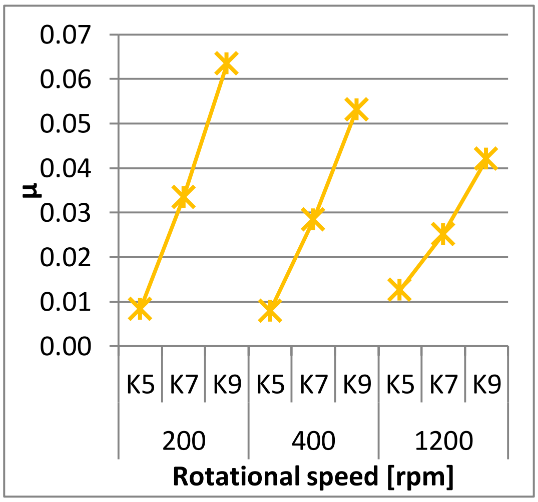

Figure 161: 501 gear design friction coefficient lubricated with ESTR oil.

A.3. Oils data sheets

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40
Gear Ref.: 2473
Side: A

Oil ref.: PAOR
Date: 25-10-2012 to 30-10-2012
Material: 20MnCr5+case hardned

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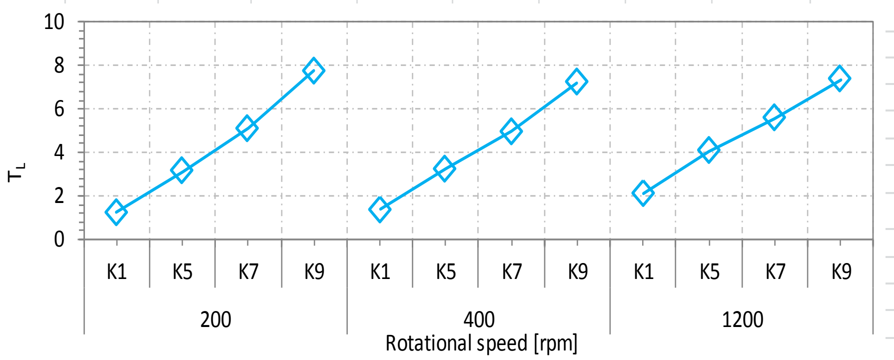
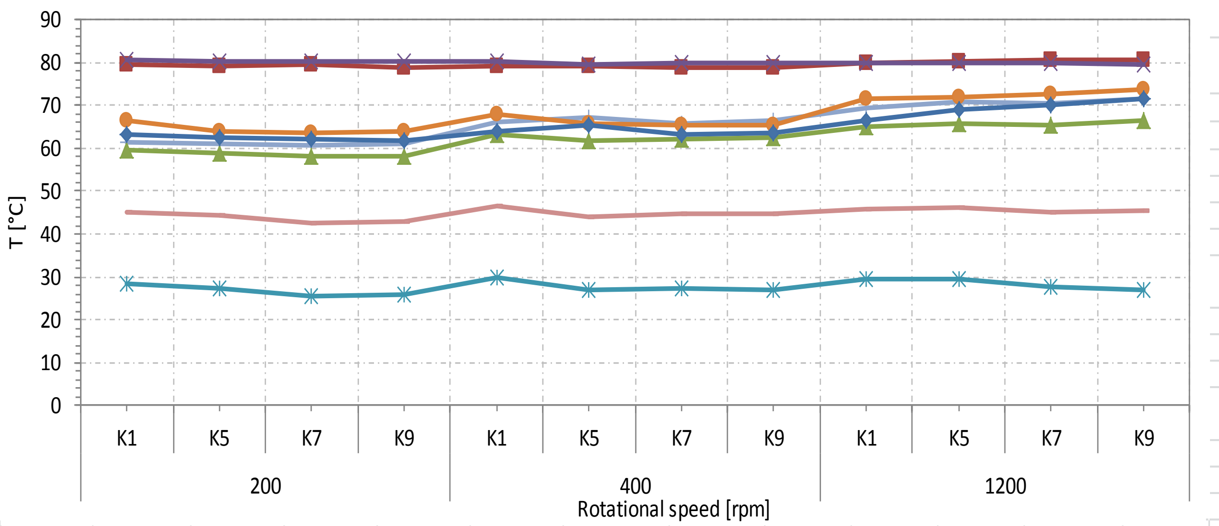
Torque Loss		
200	K1	1.22
	K5	3.12
	K7	5.08
	K9	7.73
400	K1	1.37
	K5	3.23
	K7	4.95
	K9	7.21
1200	K1	2.13
	K5	4.06
	K7	5.55
	K9	7.34





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	79.60	63.10	59.66	80.74	28.43	66.50	61.26	45.12
	K5	79.22	62.36	59.05	80.29	27.26	64.00	61.11	44.37
	K7	79.48	62.00	58.09	80.34	25.62	63.57	60.55	42.53
	K9	78.94	61.91	58.21	80.09	25.72	63.94	61.02	42.94
400	K1	79.21	64.01	63.04	80.36	29.80	67.93	66.26	46.43
	K5	79.28	65.45	61.91	79.66	26.93	65.74	67.11	43.84
	K7	78.76	63.15	62.16	79.92	27.23	65.32	65.80	44.63
	K9	78.90	63.53	62.47	79.99	27.04	65.50	66.39	44.75
1200	K1	79.92	66.35	64.92	79.90	29.49	71.51	69.51	45.79
	K5	80.15	69.19	65.79	79.78	29.62	71.95	70.69	46.24
	K7	80.48	69.94	65.55	79.92	27.81	72.48	70.55	45.17
	K9	80.66	71.49	66.33	79.69	27.10	73.68	71.44	45.28

FZG Torque loss test

TEST DATA SHEET

Instituto de Engenharia Mecânica e Gestão Industrial

CETRIB Unidade de Tribologia, Vibrações e Manutenção Industrial

Test Ref.: TL-501

Oil ref.: PAOR

Gear Ref.:

Date: 26-02-2013 to 07-03-2013

Side: A

Material: 20MnCr5+case hardned

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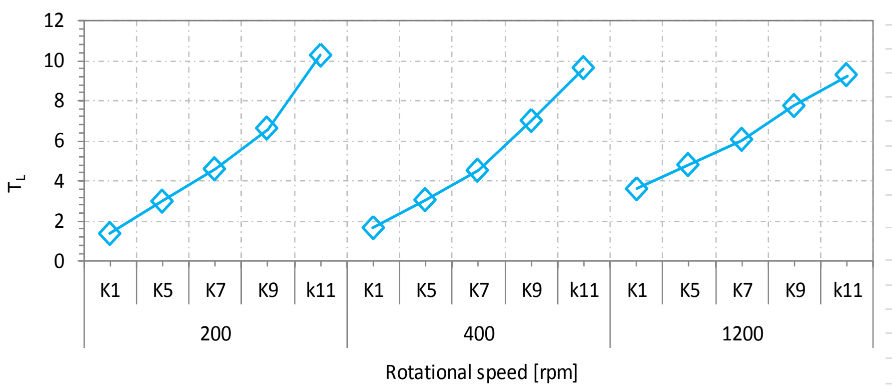
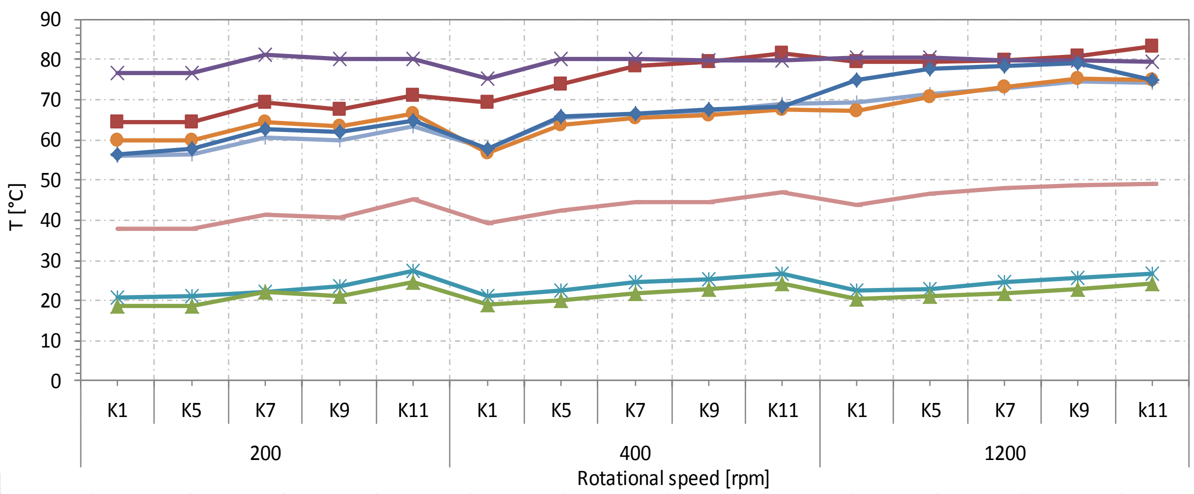
Torque Loss		
200	K1	1.38
	K5	2.96
	K7	4.58
	K9	6.59
	k11	10.28
400	K1	1.68
	K5	3.06
	K7	4.52
	K9	7.02
	k11	9.59
1200	K1	3.62
	K5	4.79
	K7	6.04
	K9	7.74
	k11	9.23





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	64.41	56.49	18.61	76.58	20.69	59.77	56.07	37.95
	K5	64.45	57.72	18.74	76.75	20.96	60.04	56.56	37.88
	K7	69.37	62.75	22.17	81.28	22.03	64.32	60.45	41.28
	K9	67.60	62.11	21.22	80.19	23.40	63.51	59.87	40.59
	K11	71.19	64.72	24.61	80.20	27.31	66.47	63.21	45.12
400	K1	69.38	57.72	19.18	75.26	21.17	56.85	57.94	39.11
	K5	73.80	65.88	19.95	80.32	22.34	63.79	65.65	42.30
	K7	78.41	66.69	21.98	80.05	24.43	65.50	66.68	44.68
	K9	79.47	67.56	22.86	79.92	25.18	66.11	67.20	44.52
	K11	81.46	68.36	24.18	79.70	26.82	67.56	68.98	47.08
1200	K1	79.38	74.94	20.41	80.51	22.49	67.30	69.35	43.90
	K5	79.61	77.73	20.96	80.37	22.94	70.69	71.35	46.69
	K7	79.98	78.47	21.90	79.98	24.74	73.01	72.66	47.97
	K9	80.92	79.16	22.75	79.97	25.48	75.43	74.63	48.81
	k11	83.13	74.85	24.36	79.35	26.68	74.82	74.12	48.96

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-951 Oil ref.: PAOR

Gear Ref.: 951/10/01 Date: 27-05-2013 to 03-06-2013

Side: A Material: 20MnCr5+case hardned

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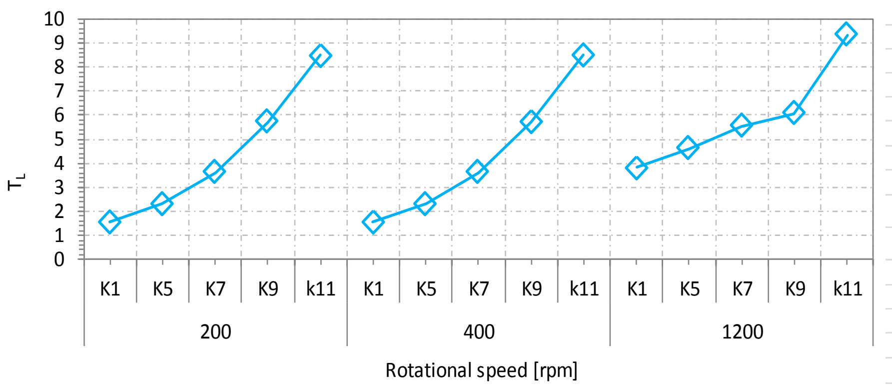
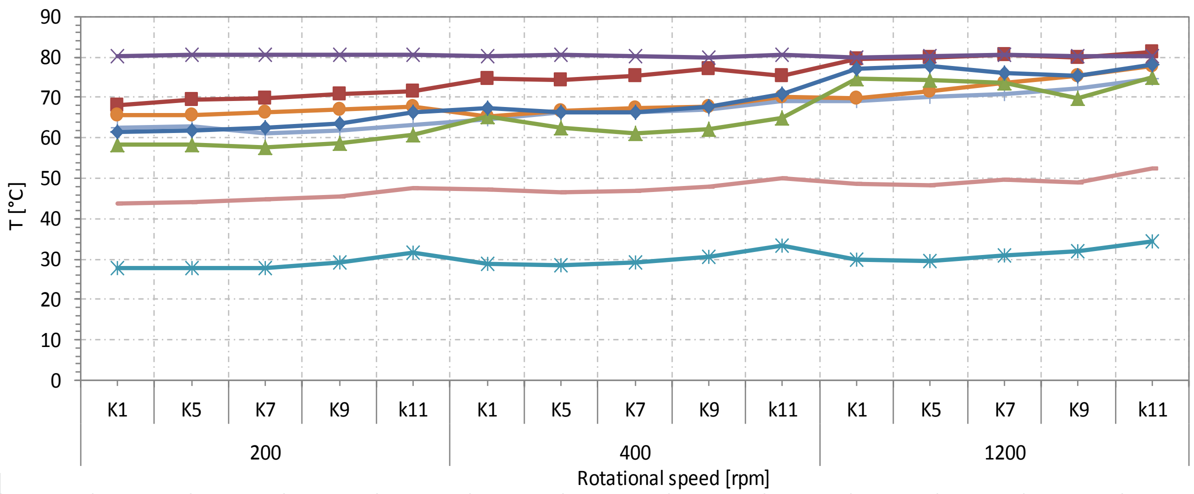
Torque Loss		
200	K1	1.56
	K5	2.31
	K7	3.63
	K9	5.74
	k11	8.48
400	K1	1.56
	K5	2.31
	K7	3.63
	K9	5.74
	k11	8.48
1200	K1	3.81
	K5	4.61
	K7	5.56
	K9	6.08
	k11	9.33





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	68.23	61.47	58.20	80.07	27.65	65.58	62.49	43.87
	K5	69.53	61.71	58.31	80.49	27.69	65.80	62.93	43.98
	K7	69.97	62.62	57.74	80.65	27.87	66.47	61.12	44.89
	K9	70.76	63.68	58.54	80.64	29.05	66.89	61.99	45.65
	k11	71.69	66.27	60.66	80.59	31.49	67.71	63.24	47.54
400	K1	74.74	67.34	65.25	80.22	28.77	65.33	64.69	47.10
	K5	74.17	66.33	62.41	80.43	28.48	66.82	66.18	46.43
	K7	75.47	66.20	61.00	80.21	29.32	67.35	66.24	46.77
	K9	77.29	67.64	62.06	80.00	30.50	67.85	67.14	47.82
	k11	75.37	70.87	64.87	80.49	33.18	70.29	69.28	50.00
1200	K1	79.65	77.07	74.81	80.07	29.72	69.85	69.04	48.46
	K5	79.94	77.70	74.30	80.25	29.55	71.43	70.32	48.31
	K7	80.60	76.15	73.80	80.56	30.94	73.77	70.75	49.66
	K9	79.99	75.23	69.72	80.35	31.82	75.38	72.13	49.06
	k11	81.14	78.22	74.95	80.33	34.38	77.97	74.63	52.34

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40
Gear Ref.: 2473
Side: A

Oil ref.: PAOM
Date: 17-12-2012 to 20-12-2012
Material: 20MnCr5+case hardned

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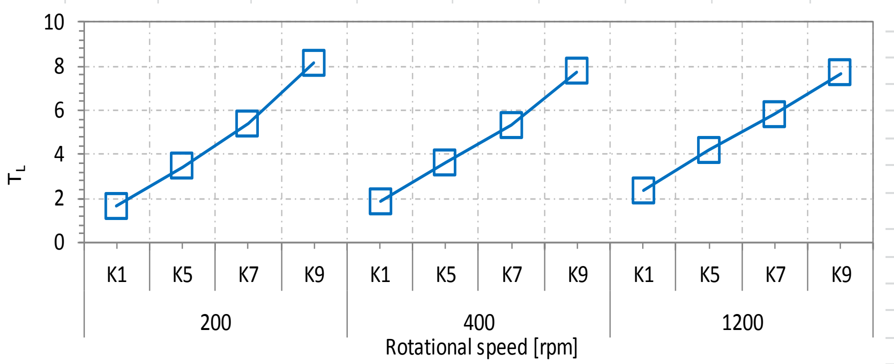
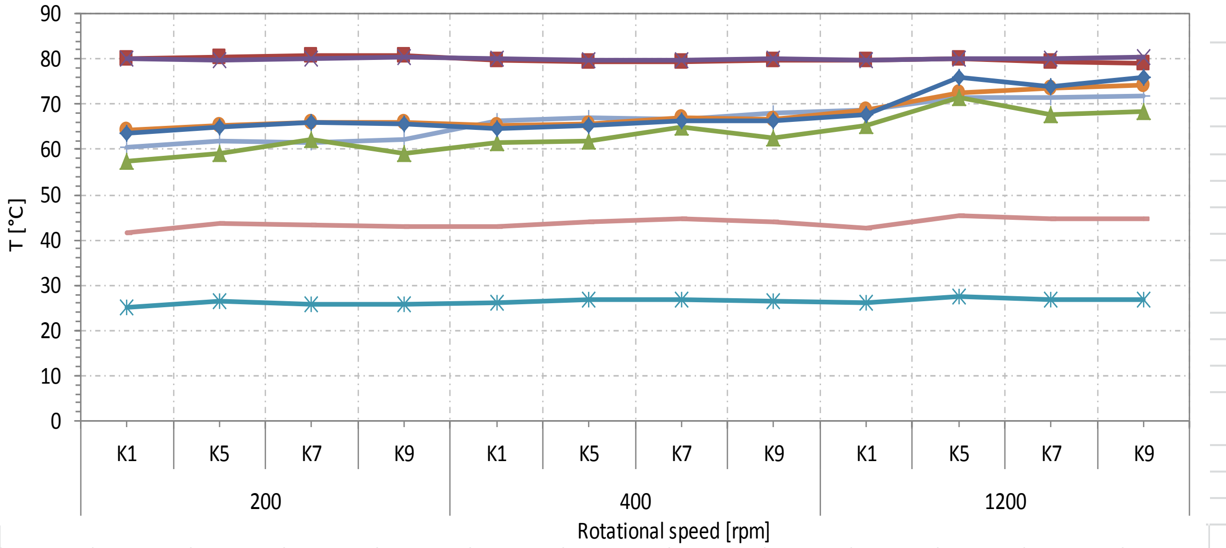
Torque Loss		
200	K1	1.63
	K5	3.45
	K7	5.37
	K9	8.15
400	K1	1.88
	K5	3.61
	K7	5.32
	K9	7.74
1200	K1	2.35
	K5	4.17
	K7	5.79
	K9	7.69





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	80.17	63.44	57.57	80.08	24.99	64.39	60.50	41.56
	K5	80.52	64.96	58.99	79.78	26.64	65.39	61.84	43.60
	K7	80.83	65.94	62.13	79.94	25.84	65.91	61.68	43.17
	K9	80.76	65.53	59.26	80.27	25.86	66.12	62.23	43.01
400	K1	79.64	64.67	61.65	80.12	25.96	65.43	66.22	42.95
	K5	79.28	65.45	61.91	79.66	26.93	65.74	67.11	43.84
	K7	79.34	66.22	65.05	79.63	26.69	66.93	66.51	44.67
	K9	79.63	66.47	62.70	80.20	26.53	66.67	68.00	44.01
1200	K1	79.71	67.70	65.25	79.91	25.98	68.78	68.73	42.76
	K5	80.06	75.96	71.65	79.96	27.50	72.44	71.48	45.45
	K7	79.43	74.00	67.70	80.18	26.91	73.47	71.37	44.57
	K9	79.19	75.85	68.43	80.43	26.70	74.40	71.89	44.85

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-501
Gear Ref.: 501/13/01
Side: B

Oil ref.: PAOM
Date: 14-02-2013 to 19-02-2013
Material: 20MnCr5+case hardned

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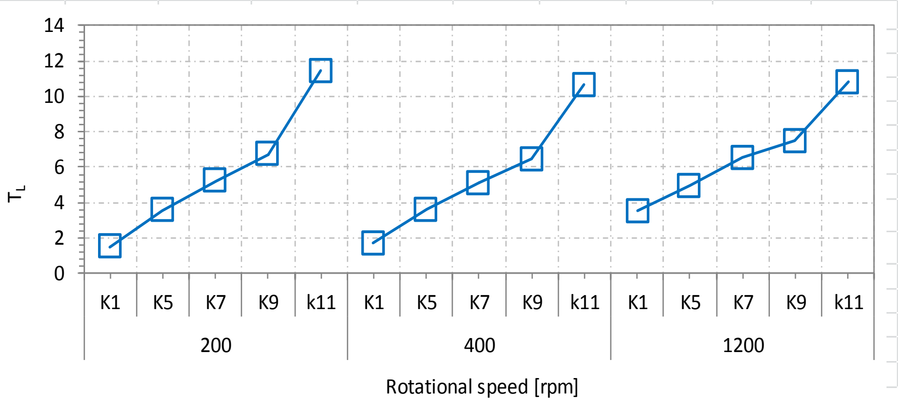
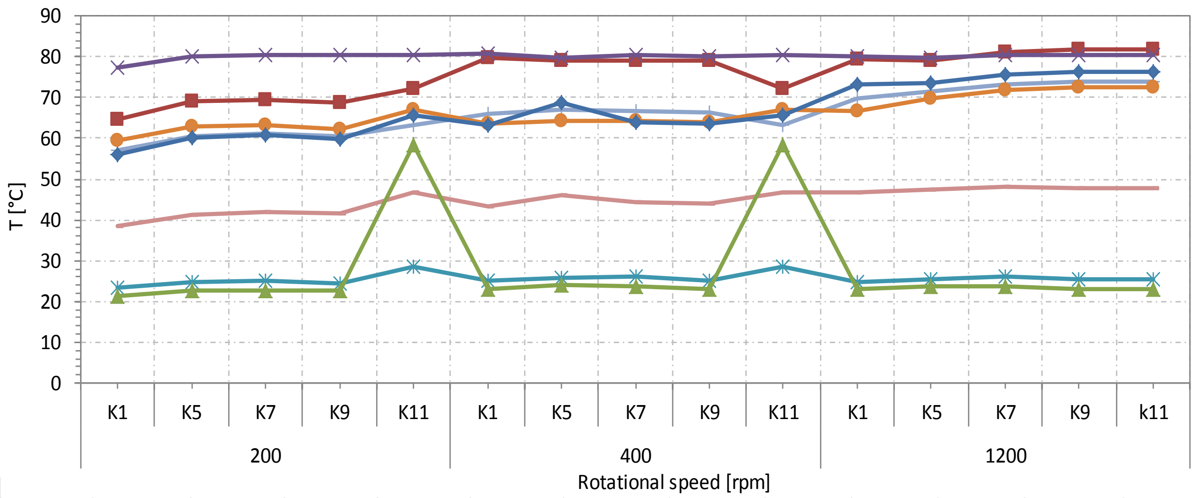
Torque Loss		
200	K1	1.49
	K5	3.56
	K7	5.24
	K9	6.73
	k11	11.44
400	K1	1.71
	K5	3.60
	K7	5.11
	K9	6.47
	k11	10.65
1200	K1	3.55
	K5	4.95
	K7	6.52
	K9	7.50
	k11	10.82





n_in	TQ_in	Test Gearbox Tout	Tbearing	Twall	Tin	Tamb	Slave Tbearing	Twall	Tbase
200	K1	64.66	56.10	21.45	77.39	23.41	59.44	57.02	38.56
	K5	68.94	60.18	22.76	80.15	24.78	62.75	60.50	41.28
	K7	69.57	60.78	22.81	80.37	25.06	63.19	61.35	42.04
	K9	68.68	59.71	22.62	80.39	24.57	62.07	60.58	41.71
	K11	72.12	65.56	58.34	80.27	28.57	67.10	63.23	46.85
400	K1	79.90	63.35	23.18	80.65	25.23	63.68	66.04	43.15
	K5	79.11	68.69	23.93	79.90	25.91	64.36	66.95	46.21
	K7	79.03	64.00	23.69	80.37	26.00	64.14	66.79	44.38
	K9	79.18	63.59	23.05	79.98	25.12	63.82	66.33	43.87
	K11	72.12	65.56	58.34	80.27	28.57	67.10	63.23	46.85
1200	K1	79.51	73.07	22.91	79.98	24.82	66.74	69.90	46.66
	K5	79.20	73.69	23.59	79.63	25.43	69.71	71.32	47.29
	K7	81.16	75.72	23.80	80.41	26.27	71.74	73.31	48.15
	K9	81.74	76.36	23.13	80.35	25.44	72.65	74.00	47.85
	k11	81.74	76.36	23.13	80.35	25.44	72.65	74.00	47.85

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40
Gear Ref.: 2473
Side: A

Oil ref.: PAOC
Date: 05-12-2012 to 10-12-2012
Material: 20MnCr5+case hardned

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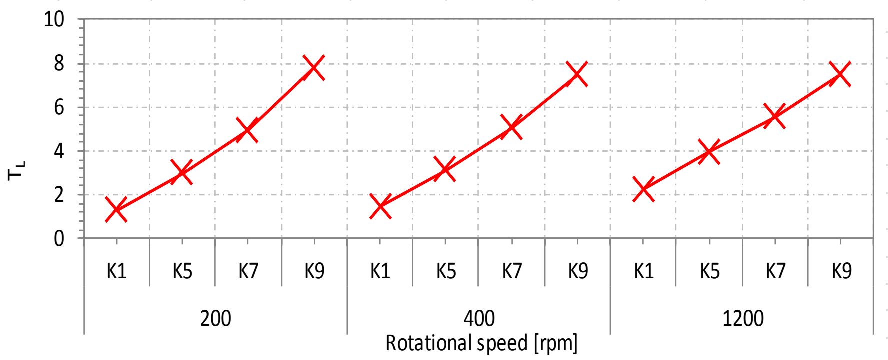
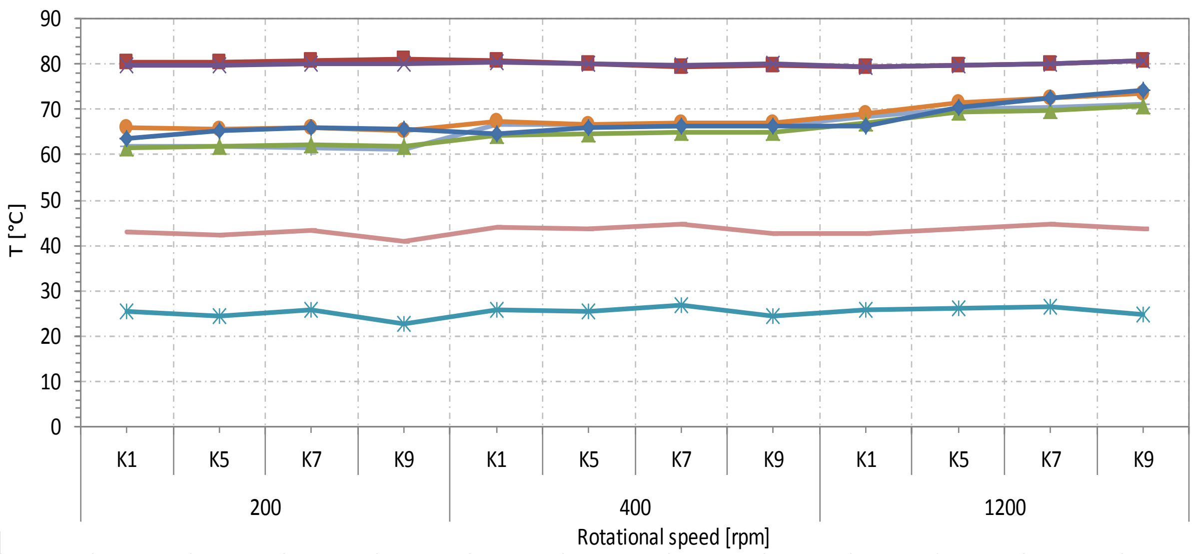
Torque Loss		
200	K1	1.27
	K5	2.98
	K7	4.94
	K9	7.78
400	K1	1.50
	K5	3.12
	K7	5.03
	K9	7.47
1200	K1	2.24
	K5	3.93
	K7	5.54
	K9	7.50





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	80.54	63.72	61.53	79.82	25.48	65.98	61.73	43.10
	K5	80.49	65.42	61.75	79.81	24.45	65.60	61.79	42.28
	K7	80.83	65.94	62.13	79.94	25.84	65.91	61.68	43.17
	K9	81.01	65.60	61.77	80.09	22.73	65.27	61.13	40.88
400	K1	80.70	64.66	64.40	80.32	25.74	67.43	66.54	43.98
	K5	80.11	66.05	64.67	80.08	25.53	66.84	66.82	43.68
	K7	79.34	66.22	65.05	79.63	26.69	66.93	66.51	44.67
	K9	79.64	66.20	64.90	80.01	24.35	66.90	66.49	42.78
1200	K1	79.55	66.40	66.90	79.53	25.81	69.07	68.44	42.59
	K5	79.91	70.63	69.31	79.87	26.30	71.47	70.19	43.61
	K7	80.01	72.38	69.82	79.93	26.64	72.68	70.32	44.68
	K9	80.62	74.36	70.74	80.69	24.83	73.61	70.98	43.80

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-501 | Oil ref.: PAOC

Gear Ref.: 501/13/01 | Date: 05-02-2013 to 08-02-2013

Side: A | Material: 20MnCr5+case hardned

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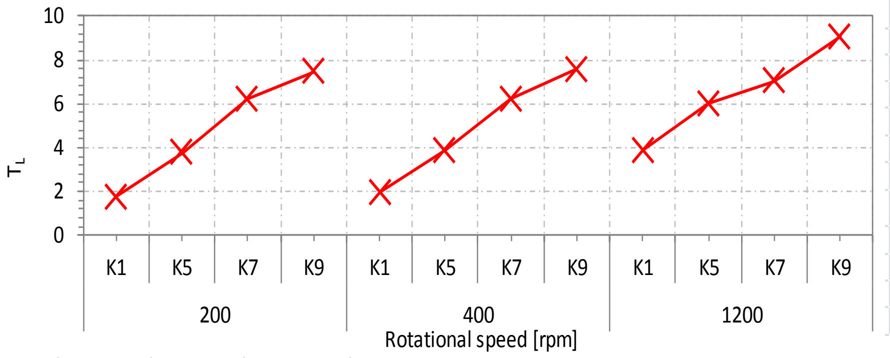
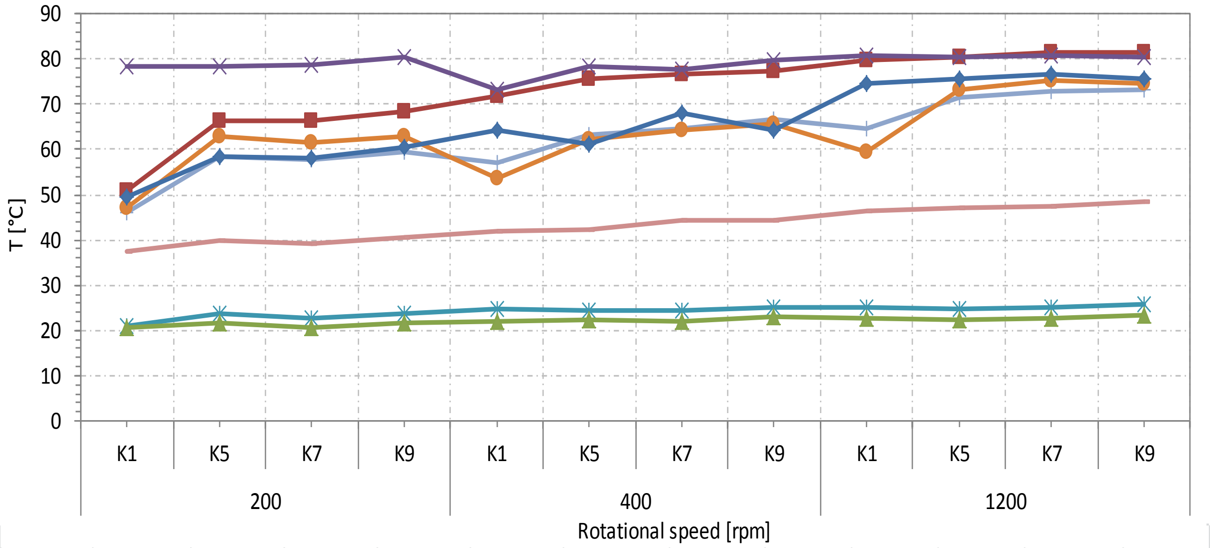
Torque Loss		
200	K1	1.74
	K5	3.75
	K7	6.20
	K9	7.44
400	K1	1.96
	K5	3.87
	K7	6.20
	K9	7.57
1200	K1	3.91
	K5	6.00
	K7	7.03
	K9	9.05





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	50.96	49.57	20.75	78.43	21.14	47.15	46.05	37.35
	K5	66.25	58.38	21.80	78.52	23.74	62.88	58.34	39.92
	K7	66.44	58.16	20.62	78.55	22.61	61.70	57.71	39.30
	K9	68.36	60.35	21.65	80.58	23.72	62.91	59.52	40.64
400	K1	71.75	64.38	21.93	73.25	24.70	53.51	57.16	41.93
	K5	75.63	61.25	22.25	78.38	24.25	62.07	63.09	42.45
	K7	76.48	68.14	22.15	77.65	24.54	64.38	64.62	44.34
	K9	77.46	64.17	22.92	79.86	25.21	65.76	66.61	44.18
1200	K1	79.89	74.50	22.59	80.76	25.20	59.55	64.55	46.35
	K5	80.47	75.69	22.43	80.51	24.69	73.33	71.64	47.02
	K7	81.28	76.49	22.69	80.80	25.04	75.14	72.80	47.49
	K9	81.47	75.65	23.27	80.25	25.75	74.45	73.06	48.47

FZG Torque loss test

TEST DATA SHEET

Instituto de Engenharia Mecânica e Gestão Industrial

CETRIB Unidade de Tribologia, Vibrações e Manutenção Industrial

Test Ref.: TL-C40

Oil ref.: PAOX

Gear Ref.: 2473

Date: 03-01-2012 to 08-01-2013

Side: A

Material: 20MnCr5+case hardned

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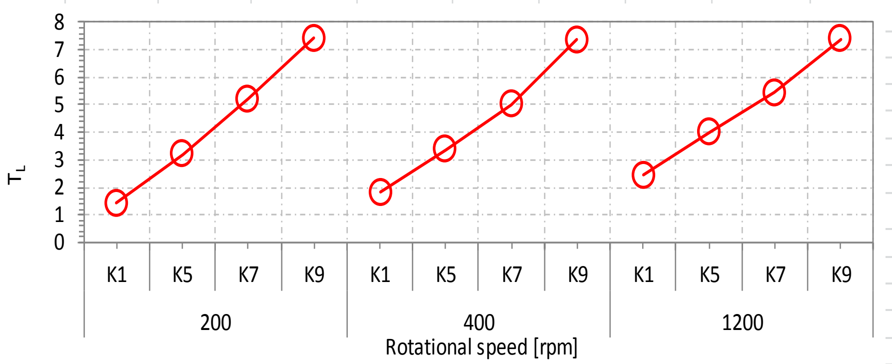
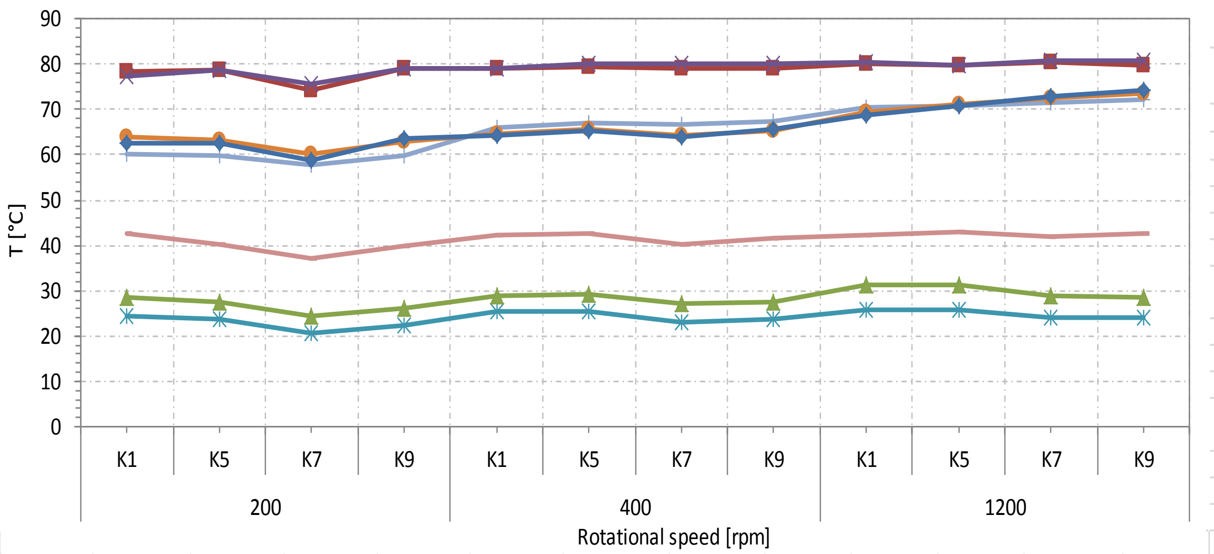
Torque Loss		
200	K1	1.43
	K5	3.19
	K7	5.20
	K9	7.42
400	K1	1.81
	K5	3.39
	K7	5.02
	K9	7.37
1200	K1	2.47
	K5	4.00
	K7	5.44
	K9	7.40





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	78.37	62.66	28.50	77.30	24.53	63.76	60.25	42.69
	K5	78.73	62.65	27.39	78.78	23.72	63.24	59.74	40.27
	K7	74.33	58.93	24.30	75.56	20.53	60.19	57.85	37.18
	K9	79.03	63.42	26.24	78.99	22.22	63.07	59.97	39.88
400	K1	79.05	64.13	28.92	79.21	25.34	64.65	66.12	42.15
	K5	79.53	65.16	29.19	80.19	25.53	65.60	67.16	42.58
	K7	79.20	64.03	27.07	80.11	23.13	64.38	66.56	40.25
	K9	79.21	65.55	27.51	80.05	23.72	65.31	67.21	41.62
1200	K1	80.20	68.64	31.41	80.27	25.74	69.27	70.31	42.44
	K5	79.59	70.93	31.37	79.85	25.88	71.11	70.78	42.83
	K7	80.28	72.75	28.84	80.67	24.16	72.37	71.57	42.01
	K9	79.87	74.37	28.47	80.82	24.08	73.47	72.01	42.66

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-501
Gear Ref.: 501/13/02
Side: A

Oil ref.: PAOX
Date: 23-01-2013 to 28-01-2013
Material: 20MnCr5+case hardned

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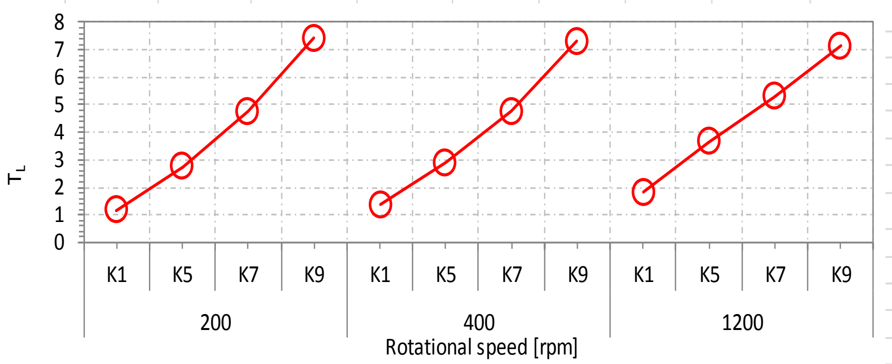
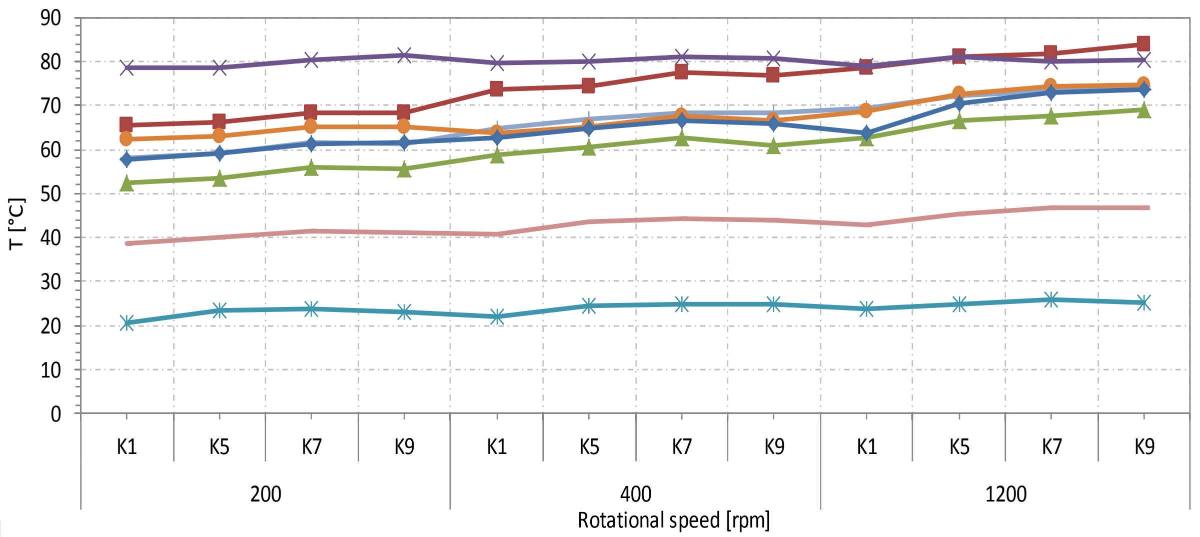
Torque Loss		
200	K1	1.16
	K5	2.75
	K7	4.77
	K9	7.42
400	K1	1.38
	K5	2.90
	K7	4.76
	K9	7.31
1200	K1	1.85
	K5	3.65
	K7	5.31
	K9	7.14





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	65.36	57.88	52.44	78.57	20.68	62.37	57.94	38.50
	K5	66.07	59.23	53.63	78.50	23.40	62.91	59.04	40.18
	K7	68.28	61.26	55.82	80.21	23.84	65.15	61.66	41.58
	K9	68.45	61.47	55.43	81.31	23.17	65.02	61.07	40.95
400	K1	73.59	62.67	58.87	79.61	21.93	63.55	64.91	40.89
	K5	74.44	64.61	60.55	80.07	24.58	65.30	66.76	43.53
	K7	77.48	66.40	62.71	80.99	24.98	67.63	68.15	44.40
	K9	76.90	65.93	60.98	80.67	24.85	66.38	68.15	43.99
1200	K1	78.56	63.80	62.81	79.01	23.83	68.62	69.29	42.78
	K5	80.88	70.45	66.59	81.03	24.73	72.48	72.25	45.37
	K7	81.83	72.93	67.70	80.08	25.84	74.24	73.11	46.83
	K9	83.76	73.65	68.99	80.49	25.17	74.76	74.46	46.85

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40

Gear Ref.: 2473

Side: A

Oil ref.: PAGD

Date: 09-11-2012 to 14-11-2012

Material: 20MnCr5+case hardned

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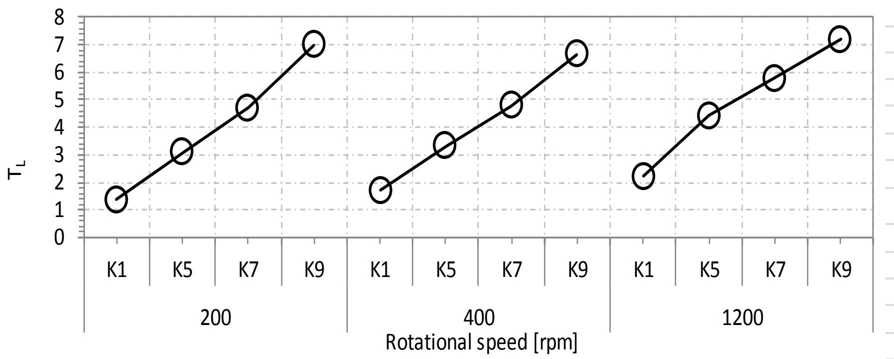
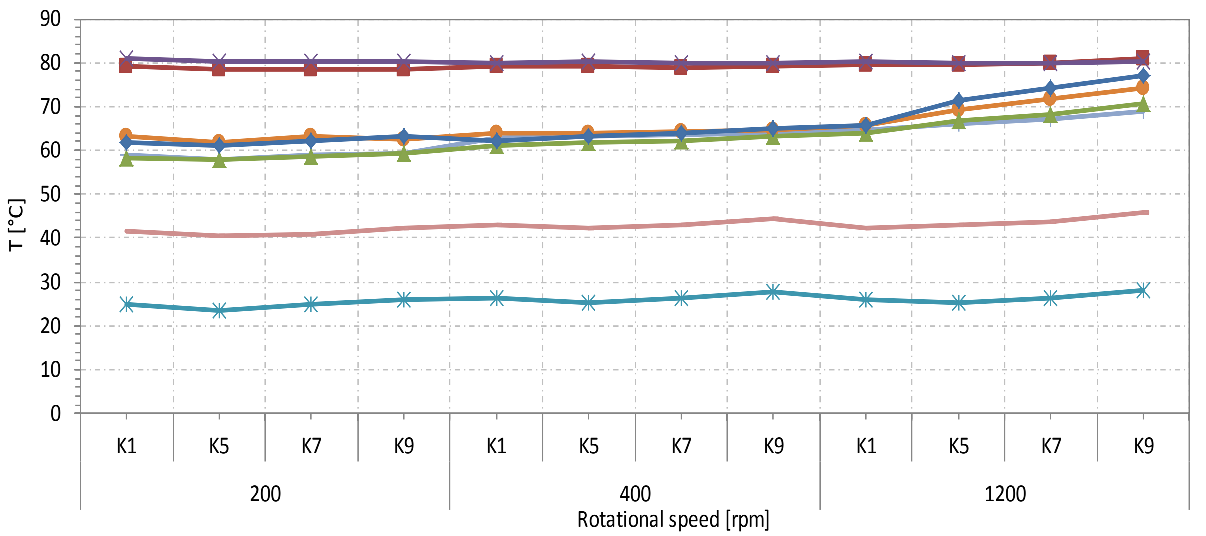
Torque Loss		
200	K1	1.38
	K5	3.08
	K7	4.70
	K9	7.00
400	K1	1.73
	K5	3.33
	K7	4.79
	K9	6.65
1200	K1	2.25
	K5	4.42
	K7	5.76
	K9	7.19





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	79.23	61.93	58.37	80.88	24.97	63.16	58.82	41.71
	K5	78.31	61.19	57.81	80.24	23.54	61.86	58.05	40.54
	K7	78.48	61.99	58.50	80.12	24.73	63.06	58.96	40.97
	K9	78.42	63.20	59.42	80.39	25.83	62.35	59.45	42.19
400	K1	79.04	62.27	61.22	80.06	26.19	63.92	62.97	43.04
	K5	79.22	63.35	61.67	80.17	25.20	63.88	63.16	42.36
	K7	78.87	63.81	62.27	79.95	26.28	64.29	63.43	43.00
	K9	79.04	65.07	63.29	79.95	27.59	64.70	64.09	44.46
1200	K1	79.67	65.70	64.08	80.13	25.99	65.74	64.57	42.33
	K5	79.63	71.27	66.89	79.95	25.36	69.23	65.98	42.86
	K7	80.07	74.09	68.21	79.95	26.15	71.87	67.22	43.86
	K9	80.85	77.23	70.69	80.11	28.03	74.15	68.74	46.00

FZG Torque loss test

TEST DATA SHEET

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CETRIB

Test Ref.: TL-501

Gear Ref.: 501/13/02

Side: B

Oil ref.: PAGD

Date: 19-03-2013 to 26-03-2013

Material: 20MnCr5+case hardned

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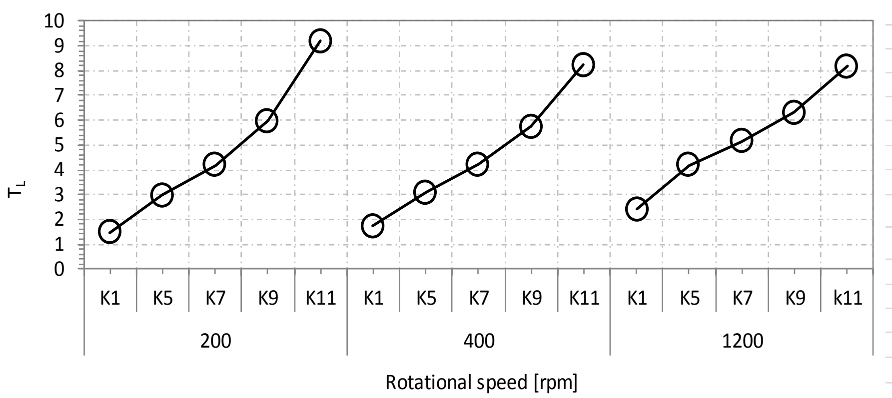
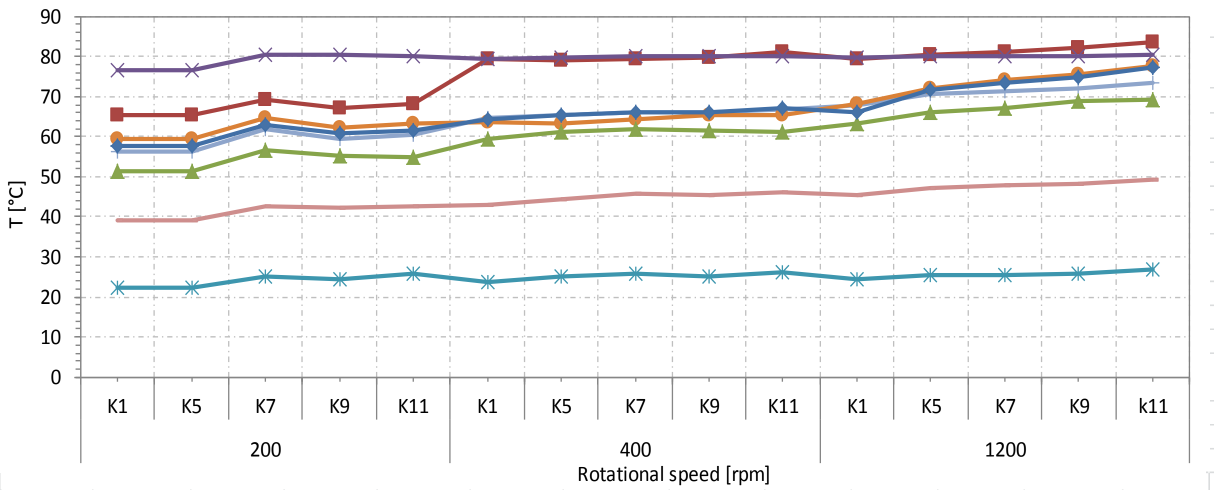
Torque Loss		
200	K1	1.48
	K5	2.96
	K7	4.19
	K9	5.97
	K11	9.19
400	K1	1.76
	K5	3.09
	K7	4.21
	K9	5.78
	K11	8.24
1200	K1	2.43
	K5	4.19
	K7	5.14
	K9	6.30
	k11	8.17





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	65.35	57.56	51.21	76.51	22.30	59.60	56.20	39.01
	K5	65.35	57.56	51.21	76.51	22.30	59.60	56.20	39.01
	K7	69.19	62.81	56.52	80.40	25.26	64.67	62.05	42.66
	K9	67.27	60.68	55.11	80.38	24.49	62.32	59.47	42.13
	K11	68.28	61.68	54.97	80.03	25.89	63.12	60.66	42.72
400	K1	79.30	64.37	59.58	79.35	23.83	63.57	64.82	43.11
	K5	79.09	65.39	61.09	79.92	24.98	63.41	65.47	44.50
	K7	79.43	65.98	61.95	80.13	25.79	64.27	66.05	45.94
	K9	79.92	65.99	61.70	79.98	25.25	65.43	66.12	45.25
	K11	81.23	67.00	61.17	80.00	26.23	65.43	66.68	46.16
1200	K1	79.24	66.01	63.38	79.79	24.36	68.17	67.68	45.31
	K5	80.47	71.71	66.13	80.05	25.56	72.18	70.50	47.16
	K7	81.30	73.53	67.24	80.27	25.56	74.16	71.42	47.82
	K9	82.22	74.76	68.97	80.26	25.87	75.50	72.10	48.17
	k11	83.67	77.21	69.25	80.55	26.68	77.65	73.55	49.19

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-951 | Oil ref.: PAGD

Gear Ref.: 951/10/01 | Date: 05-06-2013 to 12-06-2013

Side: A | Material: 20MnCr5+case hardned

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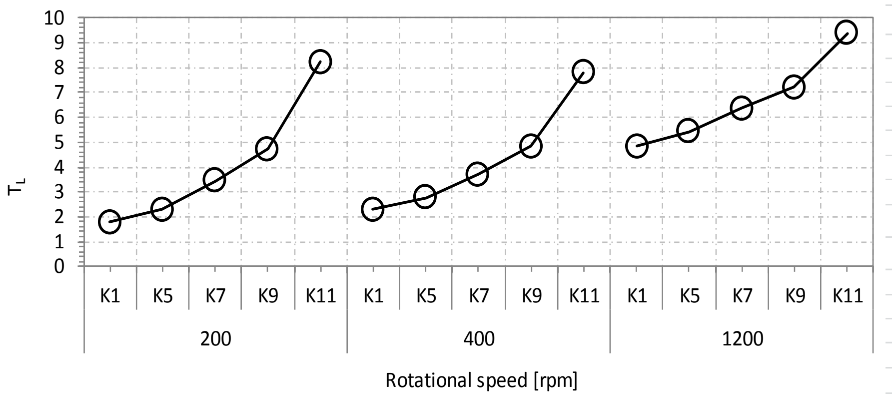
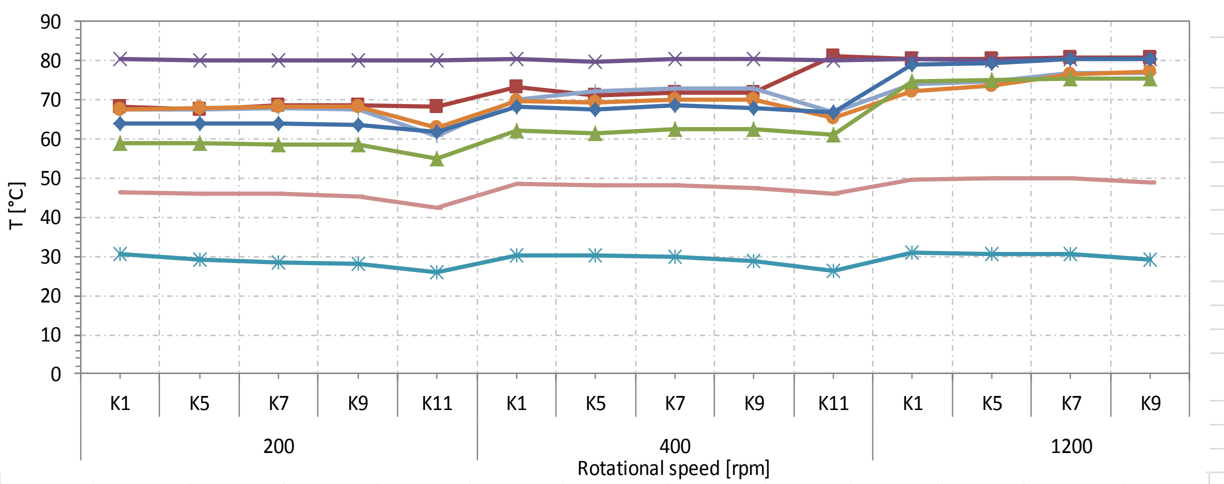
Torque Loss		
200	K1	1.77
	K5	2.29
	K7	3.45
	K9	4.73
	K11	8.25
400	K1	2.29
	K5	2.77
	K7	3.71
	K9	4.84
	K11	7.80
1200	K1	4.85
	K5	5.44
	K7	6.36
	K9	7.22
	K11	9.39





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	68.44	64.09	59.02	80.42	30.60	67.49	67.66	46.41
	K5	67.53	63.86	58.77	80.13	29.34	67.95	67.69	46.17
	K7	68.75	63.99	58.76	79.96	28.64	68.25	67.81	45.88
	K9	68.68	63.77	58.60	80.23	28.05	68.17	67.68	45.34
	K11	68.22	61.66	54.94	79.93	25.85	63.06	60.59	42.56
400	K1	73.46	68.21	62.00	80.41	30.23	69.53	70.13	48.38
	K5	71.08	67.60	61.62	79.78	30.18	69.37	72.12	48.06
	K7	71.82	68.45	62.57	80.29	29.83	70.13	72.93	48.32
	K9	71.78	68.07	62.36	80.54	28.88	70.05	72.85	47.62
	K11	81.19	67.00	61.16	79.97	26.20	65.39	66.66	46.12
1200	K1	80.40	78.95	74.86	80.48	30.93	72.24	73.93	49.79
	K5	80.29	79.29	74.97	80.19	30.68	73.58	74.78	49.91
	K7	80.64	80.28	75.41	80.34	30.58	76.63	76.71	50.09
	K9	80.83	80.33	75.32	80.40	29.34	77.18	76.83	48.99
	K11	83.65	77.19	69.22	80.48	26.72	77.64	73.55	49.14

FZG Torque loss test

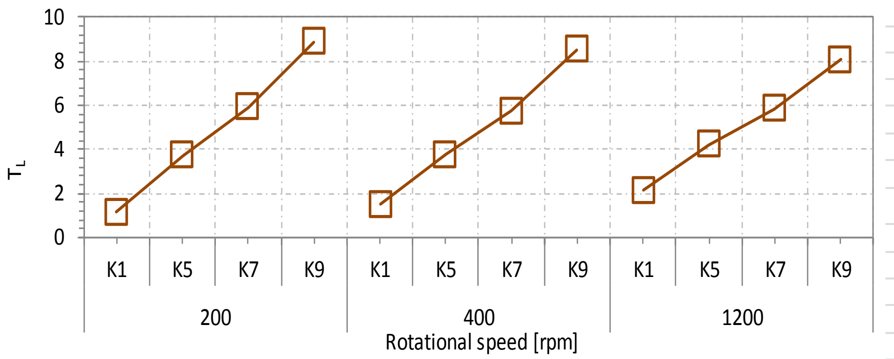
TEST DATA SHEET

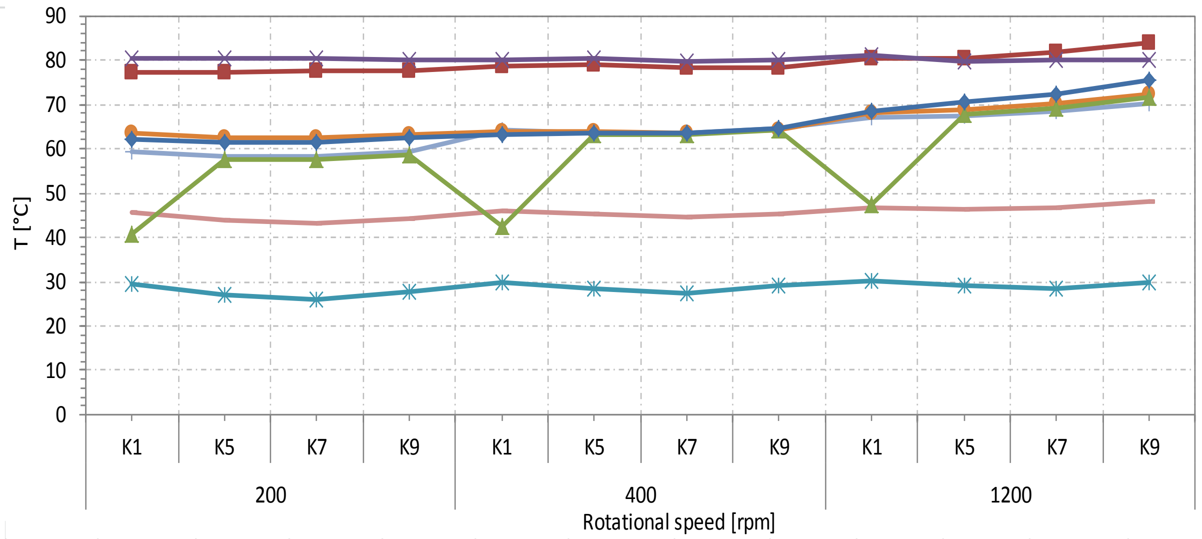
Test Ref.: TL-C40 | Oil ref.: MINR

Gear Ref.: 2473 | Date: 11-10-2012 to 16-10-2012

Side: A | Material: 20MnCr5+case hardned | Page 1/1

Torque Loss		
200	K1	1.16
	K5	3.72
	K7	5.92
	K9	8.88
400	K1	1.50
	K5	3.77
	K7	5.75
	K9	8.54
1200	K1	2.13
	K5	4.22
	K7	5.86
	K9	8.08





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	77.18	62.22	40.70	80.50	29.63	63.46	59.22	45.51
	K5	77.31	61.41	57.49	80.54	27.14	62.62	58.31	43.72
	K7	77.66	61.62	57.72	80.44	25.81	62.53	58.38	43.13
	K9	77.66	62.41	58.51	80.23	27.55	63.09	59.16	44.20
400	K1	78.55	63.15	42.54	80.01	29.95	63.94	64.21	46.16
	K5	78.89	63.49	63.20	80.45	28.52	63.75	63.72	45.19
	K7	78.47	63.54	63.20	79.84	27.48	63.58	63.62	44.65
	K9	78.37	64.54	64.20	79.92	29.21	64.27	64.53	45.35
1200	K1	80.42	68.64	47.47	81.11	30.23	68.00	66.92	46.79
	K5	80.50	70.52	67.78	79.78	29.16	68.77	67.40	46.36
	K7	81.84	72.41	69.10	79.94	28.46	70.26	68.45	46.69
	K9	83.85	75.65	71.47	80.06	29.91	72.19	70.39	48.07

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-501
Gear Ref.: 501/13/02
Side: B

Oil ref.: MINR
Date: 11-03-2013 to 15-03-2013
Material: 20MnCr5+case hardned

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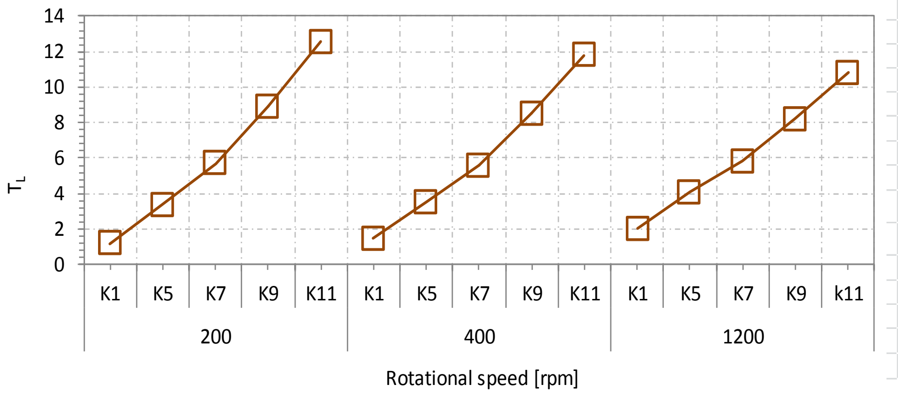
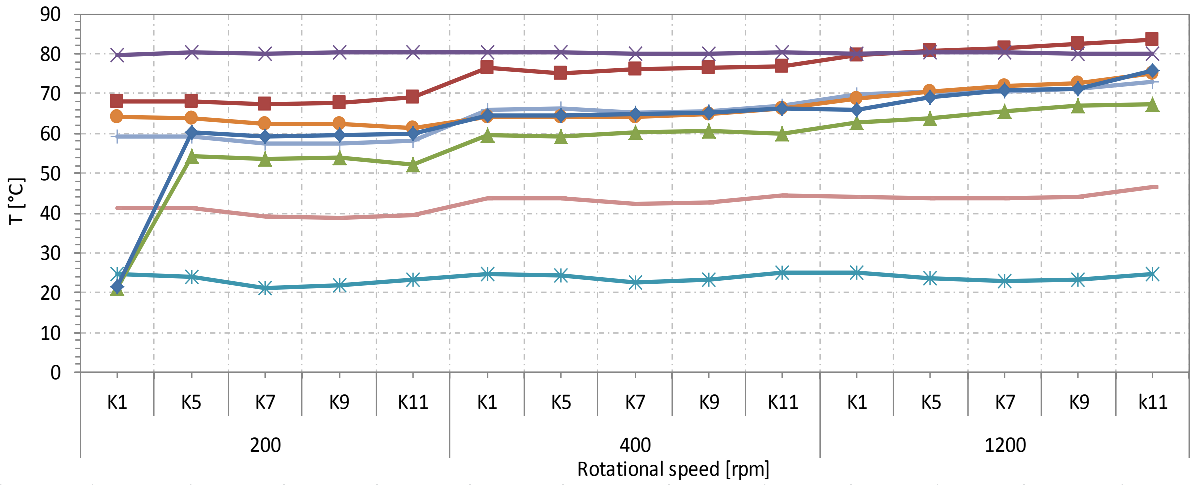
Torque Loss		
200	K1	1.17
	K5	3.35
	K7	5.71
	K9	8.90
	K11	12.57
400	K1	1.46
	K5	3.51
	K7	5.57
	K9	8.51
	K11	11.81
1200	K1	2.05
	K5	4.07
	K7	5.80
	K9	8.21
	k11	10.81





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	68.05	21.57	21.21	79.70	24.81	64.05	59.07	41.19
	K5	67.90	60.45	54.30	80.40	24.06	63.64	59.36	41.34
	K7	67.34	59.30	53.42	79.97	21.08	62.24	57.31	39.27
	K9	67.67	59.52	53.79	80.29	21.83	62.39	57.46	38.93
	K11	69.25	59.85	52.35	80.28	23.46	61.27	58.30	39.65
400	K1	76.46	64.49	59.45	80.22	24.77	64.06	65.84	43.76
	K5	75.23	64.56	59.37	80.36	24.37	64.12	66.26	43.61
	K7	76.05	64.77	60.16	80.11	22.54	64.21	65.07	42.38
	K9	76.56	65.09	60.53	79.93	23.24	64.77	65.54	42.74
	K11	76.96	66.27	59.79	80.24	24.98	66.35	66.94	44.42
1200	K1	79.53	65.84	62.79	80.13	25.10	68.74	69.77	44.03
	K5	80.68	68.93	63.90	80.22	23.53	70.58	70.59	43.86
	K7	81.47	70.91	65.59	80.27	23.08	71.74	70.55	43.88
	K9	82.55	71.17	66.84	80.06	23.42	72.68	71.15	43.90
	k11	83.50	75.70	67.40	79.85	24.65	75.16	73.09	46.37

FZG Torque loss test

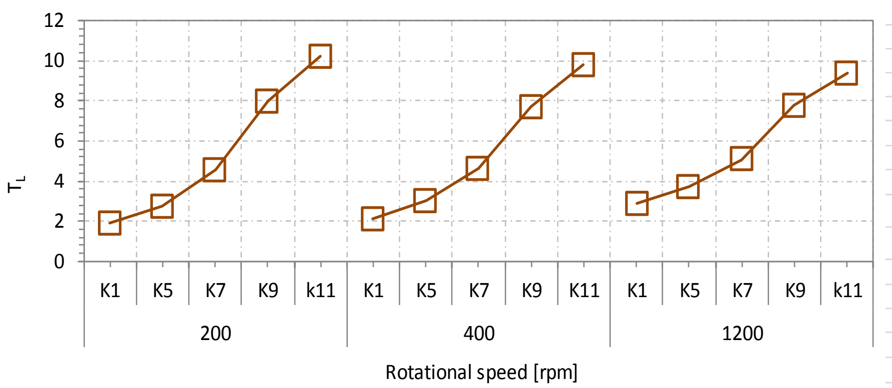
TEST DATA SHEET

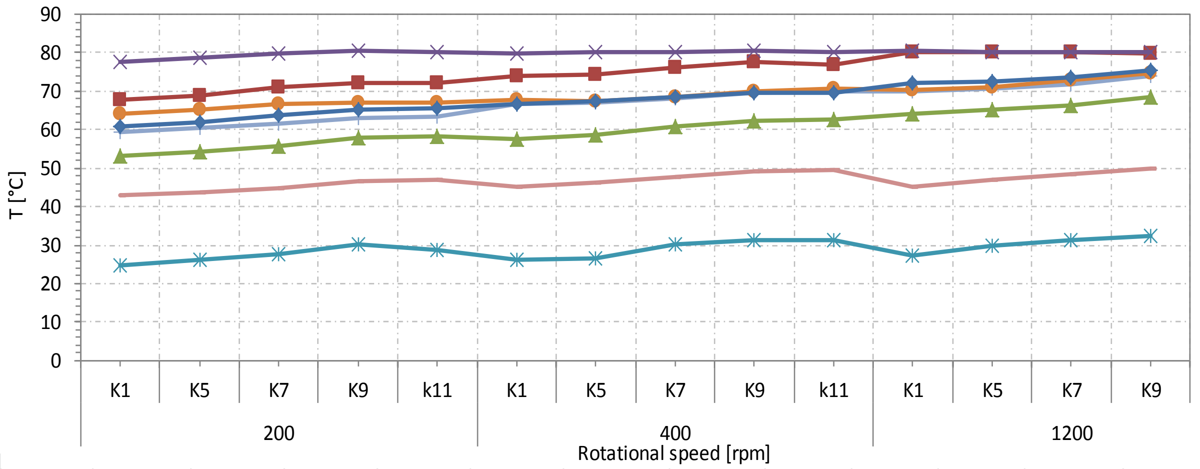
Test Ref.: TL-951 | Oil ref.: MINR

Gear Ref.: 951/10/01 | Date: 17-05-2013 to 23-05-2013

Side: A | Material: 20MnCr5+case hardned | Page 1/1

Torque Loss		
200	K1	1.89
	K5	2.74
	K7	4.54
	K9	7.96
	k11	10.20
400	K1	2.10
	K5	3.01
	K7	4.62
	K9	7.74
	K11	9.79
1200	K1	2.88
	K5	3.72
	K7	5.07
	K9	7.75
	k11	9.38





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	67.90	60.94	53.28	77.71	24.72	64.29	59.23	42.83
	K5	68.97	62.05	54.40	78.88	26.05	65.39	60.61	43.64
	K7	70.95	63.62	55.82	79.86	27.53	66.61	61.67	44.91
	K9	72.20	65.04	57.76	80.36	30.19	67.09	62.99	46.68
	k11	72.12	65.56	58.34	80.27	28.57	67.10	63.23	46.85
400	K1	73.85	66.65	57.70	79.82	26.15	67.71	66.65	45.26
	K5	74.20	67.27	58.69	80.32	26.43	67.33	67.09	46.28
	K7	76.10	68.34	60.72	80.12	30.13	68.37	68.08	47.57
	K9	77.63	69.54	62.18	80.39	31.34	69.78	69.51	49.23
	k11	76.91	69.63	62.72	80.18	31.33	70.68	70.05	49.60
1200	K1	80.15	72.23	64.10	80.38	27.25	70.34	69.83	45.30
	K5	80.10	72.61	65.08	80.23	29.97	71.19	70.51	46.85
	K7	80.08	73.60	66.26	80.24	31.31	72.93	71.82	48.38
	K9	79.90	75.26	68.45	80.29	32.55	74.76	74.14	50.07
	k11	79.58	75.84	68.92	80.11	31.85	76.45	74.99	50.06

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40
Gear Ref.: 2473
Side: A

Oil ref.: MINE
Date: 18-10-2012 to 23-10-2012
Material: 20MnCr5+case hardned

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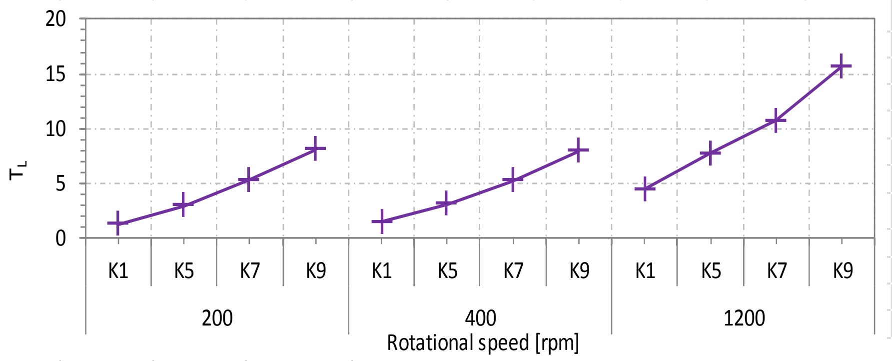
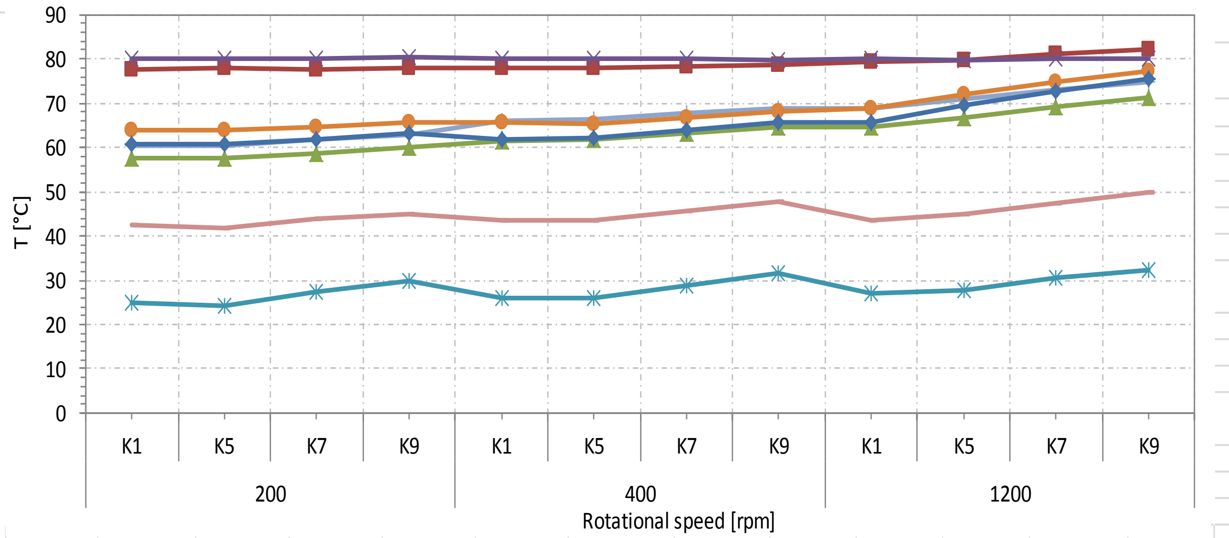
Torque Loss		
200	K1	1.24
	K5	2.97
	K7	5.32
	K9	8.08
400	K1	1.56
	K5	3.14
	K7	5.27
	K9	7.93
1200	K1	4.44
	K5	7.75
	K7	10.70
	K9	15.69





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	77.65	60.63	57.60	80.17	24.83	64.04	60.45	42.56
	K5	77.83	60.63	57.65	80.10	24.22	63.77	60.45	41.77
	K7	77.72	61.70	58.73	79.96	27.20	64.52	61.67	43.73
	K9	78.07	63.16	59.98	80.27	29.79	65.62	62.90	45.05
400	K1	77.97	61.84	61.41	80.18	25.83	65.51	65.90	43.41
	K5	77.96	62.22	61.67	80.05	26.09	65.47	66.23	43.43
	K7	78.48	63.91	63.09	80.12	28.92	66.80	67.64	45.51
	K9	78.66	65.60	64.45	79.77	31.62	67.97	68.71	47.80
1200	K1	79.49	65.79	64.65	80.19	26.88	68.74	68.81	43.55
	K5	79.73	69.38	66.60	79.67	27.55	71.82	70.90	44.91
	K7	81.19	72.68	69.19	80.04	30.64	74.93	73.09	47.53
	K9	82.13	75.40	71.40	79.94	32.14	77.35	74.87	49.71

FZG Torque loss test

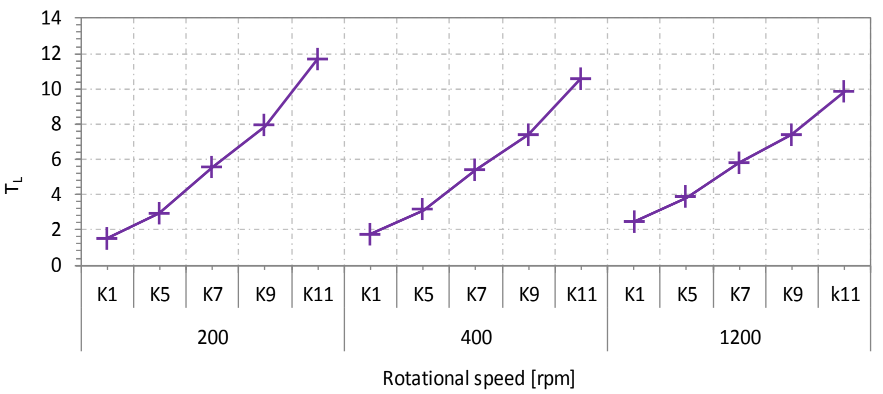
TEST DATA SHEET

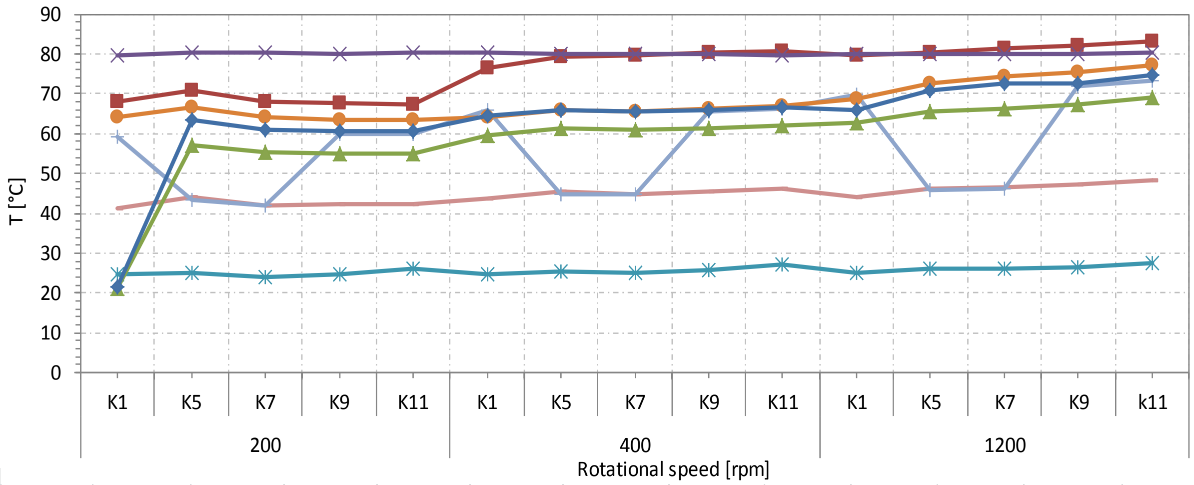
Institute de Engenharia Mecânica e Gestão Industrial

CETRIB

Test Ref.: TL-501 | Oil ref.: MINE
Gear Ref.: 501/13/02 | Date: 09-04-2013 to 15-04-2013
Side: B | Material: 20MnCr5+case hardned | Page 1/1

Torque Loss		
200	K1	1.49
	K5	2.93
	K7	5.54
	K9	7.88
	K11	11.73
400	K1	1.74
	K5	3.13
	K7	5.38
	K9	7.38
	K11	10.57
1200	K1	2.44
	K5	3.86
	K7	5.79
	K9	7.39
	k11	9.83





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	68.05	21.57	21.21	79.70	24.81	64.05	59.07	41.19
	K5	70.80	63.40	57.03	80.30	24.86	66.67	43.44	44.19
	K7	68.18	60.93	55.19	80.36	23.89	64.07	41.95	42.07
	K9	67.70	60.76	54.82	79.99	24.81	63.50	59.78	42.41
	K11	67.38	60.69	55.02	80.22	25.94	63.51	60.04	42.18
400	K1	76.46	64.49	59.45	80.22	24.77	64.06	65.84	43.76
	K5	79.34	65.99	61.22	79.96	25.44	66.07	44.81	45.45
	K7	79.68	65.67	60.85	79.98	25.10	65.63	44.74	44.74
	K9	80.24	66.07	61.44	80.03	25.73	66.22	65.62	45.34
	K11	80.58	66.47	62.13	79.79	27.19	66.86	66.21	46.06
1200	K1	79.53	65.84	62.79	80.13	25.10	68.74	69.77	44.03
	K5	80.30	70.68	65.42	80.08	26.06	72.67	45.90	46.14
	K7	81.38	72.54	66.22	80.05	26.11	74.39	46.20	46.59
	K9	82.10	72.66	67.17	79.87	26.52	75.50	71.99	47.31
	k11	83.23	74.82	69.14	80.29	27.47	77.30	73.26	48.24

FZG Torque loss test

TEST DATA SHEET

Test Ref.: TL-C40

Gear Ref.: 2473

Side: A

Oil ref.: ESTR

Date: 02-11-2012 to 08-11-2012

Material: 20MnCr5+case hardned

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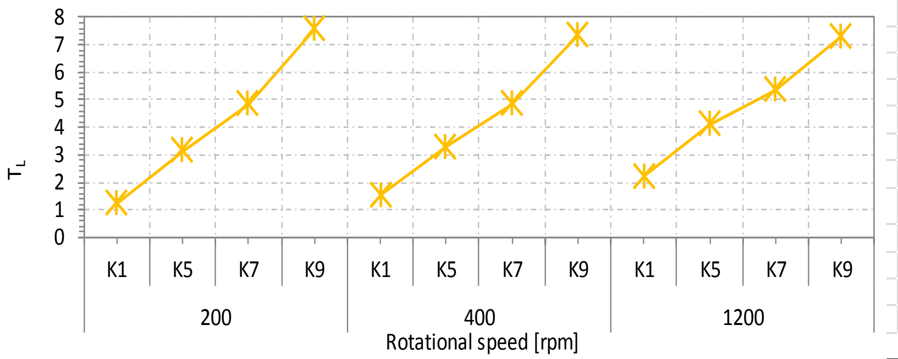
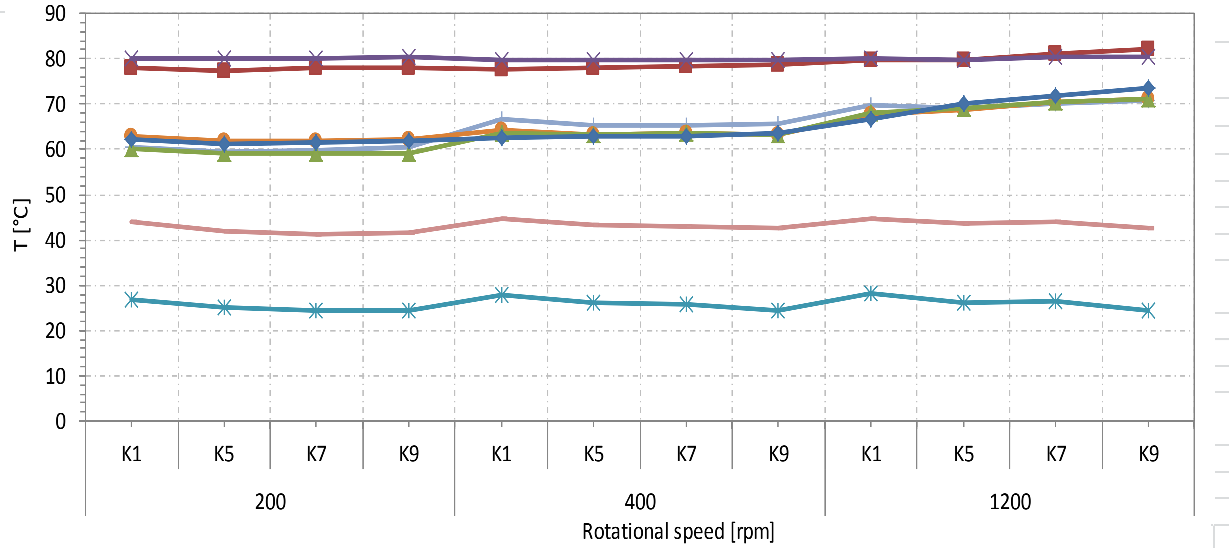
Torque Loss		
200	K1	1.24
	K5	3.14
	K7	4.85
	K9	7.56
400	K1	1.53
	K5	3.30
	K7	4.85
	K9	7.33
1200	K1	2.21
	K5	4.11
	K7	5.36
	K9	7.31





n_in	TQ_in	Test Gearbox Tout	Tbearing	Twall	Tin	Tamb	Slave Tbearing	Twall	Tbase
200	K1	78.18	62.07	60.20	80.23	26.80	62.78	60.48	43.83
	K5	77.47	61.34	59.17	79.96	25.21	61.83	59.60	42.01
	K7	78.07	61.40	59.13	80.22	24.29	61.78	59.73	41.10
	K9	78.16	61.78	59.20	80.40	24.44	62.11	60.56	41.49
400	K1	77.83	62.67	63.68	79.84	28.02	64.42	66.64	44.59
	K5	78.15	62.91	63.29	79.87	26.23	63.29	65.30	43.24
	K7	78.43	63.04	63.42	79.82	25.86	63.43	65.44	42.98
	K9	78.76	63.49	63.36	79.82	24.58	63.19	65.64	42.50
1200	K1	79.58	66.71	68.20	80.02	28.35	67.75	69.88	44.55
	K5	79.86	70.13	69.02	79.70	26.27	68.70	69.01	43.57
	K7	80.95	71.97	70.38	80.27	26.33	70.46	70.23	44.11
	K9	82.18	73.46	71.22	80.34	24.26	71.06	70.80	42.76

FZG Torque loss test

TEST DATA SHEET

Instituto de Engenharia Mecânica e Gestão Industrial

CETRIB

Test Ref.: TL-501

Gear Ref.: 501/13/02

Side: B

Oil ref.: ESTR

Date: 30-04-2013 to 02-05-2013

Material: 20MnCr5+case hardned

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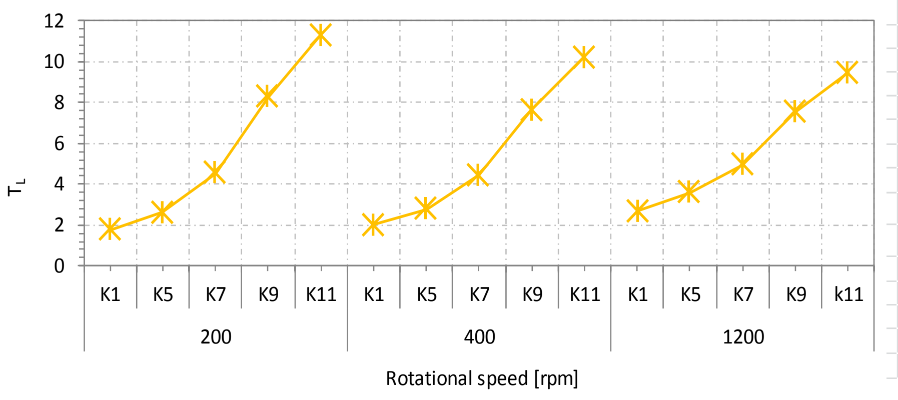
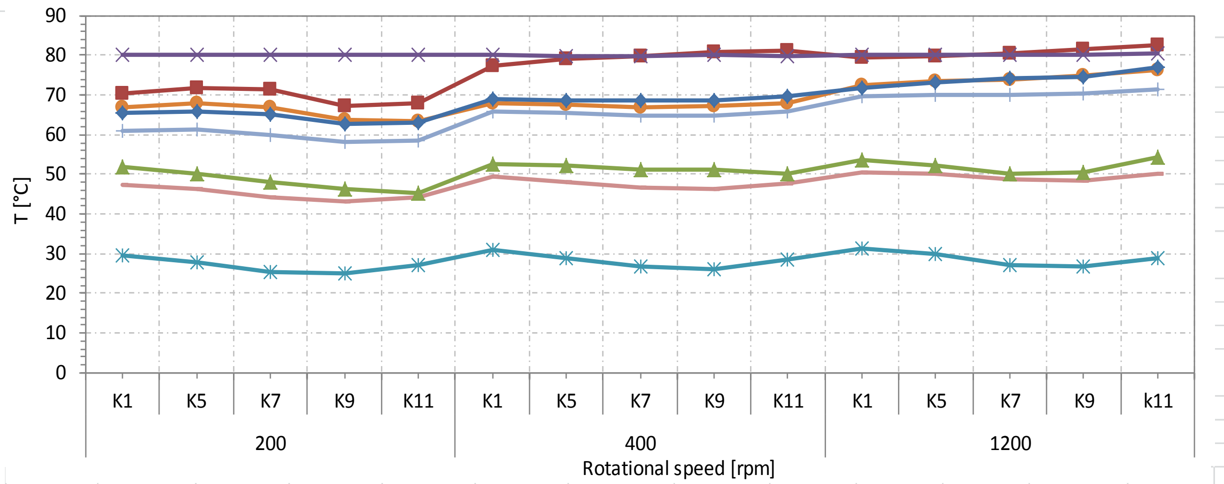
Torque Loss		
200	K1	1.77
	K5	2.62
	K7	4.53
	K9	8.27
	K11	11.33
400	K1	2.04
	K5	2.79
	K7	4.46
	K9	7.64
	K11	10.22
1200	K1	2.68
	K5	3.61
	K7	4.97
	K9	7.52
	k11	9.46





		Test Gearbox					Slave		
n_in	TQ_in	Tout	Tbearing	Twall	Tin	Tamb	Tbearing	Twall	Tbase
200	K1	70.43	65.49	52.06	80.17	29.66	66.91	60.81	47.46
	K5	71.85	66.04	50.07	80.11	27.78	67.98	61.32	46.38
	K7	71.36	65.26	48.14	80.08	25.45	66.77	60.03	44.31
	K9	67.40	62.55	46.48	80.18	25.17	63.61	58.17	43.18
	K11	67.91	62.96	45.38	80.08	27.01	63.54	58.59	44.10
400	K1	77.51	68.84	52.71	80.07	30.82	67.90	65.96	49.34
	K5	79.22	68.65	52.17	79.76	28.79	67.53	65.48	48.17
	K7	79.88	68.60	51.22	79.92	26.92	67.06	64.82	46.71
	K9	80.73	68.81	51.11	80.20	26.23	67.29	64.98	46.25
	K11	81.39	69.54	50.13	79.99	28.52	68.09	65.85	47.84
1200	K1	79.57	71.90	53.54	80.20	31.34	72.60	69.84	50.56
	K5	79.74	73.03	52.14	80.07	29.95	73.38	70.22	50.23
	K7	80.44	74.24	50.28	80.09	27.12	73.82	69.96	48.68
	K9	81.51	74.76	50.61	80.32	26.85	74.96	70.31	48.52
	k11	82.62	77.01	54.41	80.39	28.96	76.48	71.28	50.30